MECHANICAL LABORATORY METHODS

THE TESTING OF INSTRUMENTS AND MACHINES IN THE MECHANICAL ENGINEERING LABORATORY AND IN PRACTICE

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112 ILLUSTRATIONS

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PREFACE TO THIRD EDITION.

The rendering of a new edition of this work has appeared necessary in view of the latest standards of performance now being developed by the Power Test Committees of the American Society of Mechanical Engineers. Of these committees, the one on Definitions and Values has produced a Code of most general interest, covering, as it does, units of capacity and efficiency of all the prime movers dealt with in more detail by the other Codes. The Definitions and Values Code is therefore included in this edition as a second Appendix, and references made to it in the text.

Advantage has been taken to make various other changes and additions, dealing with test apparatus and methods of calculation. The section on Combustion has been entirely rewritten in the hope to make easier this difficult subject.

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Johns Hopkins University,
July, 1922.

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MECHANICAL LABORATORY METHODS

INTRODUCTION

THE PRINCIPLES OF EXPERIMENTAL MEASUREMENTS

THE science of experimental engineering rests primarily upon the art of measurement. Conclusions relating to general laws or specific operating conditions, formed from test results, stand or fall according to the accuracy of measurement.

Absolute Accuracy. There is no such thing as absolute accuracy of measurement except by accident. For instance, the length of a bar, measured with a scale graduated to one-hundredths of an inch, may appear to be 3 ins. By chance, it may be that the bar is 3 ins. long exactly, neither more nor less by an infinitesimal fraction. This, however, is very improbable. A micrometer may show it to be 3.005 ins. long. An instrument of still nicer capability might yield another decimal place, and so on. Thus, absolute accuracy presupposes an instrument with graduations corresponding to infinitely small fractions of a unit.

Assuming that this condition could be sensibly complied with, it would still be necessary to prove the correctness of the instrument within its own graduations, and this could only be done by comparison with absolute standard lengths, that is, material objects known to have definite linear dimensions. But this implies measurement by an absolutely accurate instru-

ment. Thus, to create such an instrument, it is necessary to have the very article desired.

Accidental Errors. Aside from instrumental limitations, accidental causes of error are a sufficient bar to absolute accuracy. Accidental errors are all those that cannot be eliminated by instrumental or other corrections. They are due entirely to chance and are not systematic except according to the law of probability and chance. Such errors are incurred in the calipering of a bar, for example, when the calipers are held on a slant instead of on a true diameter, when the calipers are held with varying degrees of pressure, when temperature changes momentarily affect the instrument, and so forth.

Accidental errors are equally likely to be positive or negative from which it follows that in a series of observations upon the same quantity the probabilities are that the errors of excess will equal those of deficiency.

Personal Errors. There is another class of errors known as "personal errors" whereby a particular observer, by reason of his character, habitually reads too high or too low. This error may be estimated scientifically, the result being what is known as the "personal equation." Generally, in mechanical experimentation, this quantity is not an important one, being small in comparison with the usually unavoidable errors.

We have, then, to consider two classes of errors which are bars to absolute accuracy in any measurement, namely, instrumental and accidental. From an engineering standpoint, however, absolute accuracy is neither necessary nor desirable, for it would be too laborious and cumbersome. It is not worth while to produce a result with ten or twenty significant figures when three or four are enough for practical requirements. Only that degree of accuracy is sought which accords with the dictates of common sense.

Per Cent of Error. When a result is represented by three significant figures the error occurring through the omission of

the fourth is less than one half of one per cent. Thus, the quantities 1010, 101.0, etc., may be correctly expressed 1015, 101.5, etc., but in each case the error is less than 5 parts in 1000 or one half of one per cent. With the higher digits the per cent error is correspondingly less, as when 999.0 is used for 999.5, the error then being about 0.05 per cent. *In most cases we cannot secure results with less than* 1 *per cent of error, so that three significant figures are ample for their presentation.* In many cases even less accuracy than this is consistent with the object or the conditions of the test.

The Precision and Accuracy of Instruments should be examined before using them for experimental measurements. Precision has reference to the fineness of graduations, accuracy to their correctness. The value of the smallest graduation of an instrument is called its "least count." Suppose, as an illustration, we wished to weigh separately a number of objects of approximately 100 lbs. so as to secure less than 1 per cent of error, that is, an error of less than 1 lb. A scales with a least count of 2 lbs., would do since a half a graduation could be estimated by eye with less probable error than 1 lb. A scales of any less precision however, would not be adequate.

Standards. The accuracy of an instrument is established only by comparison with some "standard unit," itself a copy, or a copy of a copy, of the "primary standard" kept at Washington. The details of this sort of testing are given in Part I, but it is well to emphasize here that the necessary closeness of the approximation of any standard used to the primary standard depends entirely upon the least count of the instrument to be tested. Generally the standard is sufficiently accurate if it is correct within one quarter of this least count, and, if the standard is an instrument, its least count need not be less than one-half that of the instrument tested. It is thus seen that the word standard must be accepted with a relative sense, and that for engineering purposes secondary standards may be improvised

which are just as useful as the most accurate standard obtainable.

Assuming instrumental correctness as great as consistent with the least count, there are two ways of securing greater accuracy of measured results: first, by using an instrument of finer graduations, and second, by making numerous repetitions of the measurement. When the latter is done a result with less probable error is obtained from the average of the observations than from any one of them. This is because by so doing the accidental errors previously mentioned are to some extent eliminated.

Intrinsic Evidence of Accuracy. In experimental engineering one of the greatest aids to accuracy is repetition. It is only by repeated trials that accurate conclusions can be framed because, not only are there variations due to erroneous measurement, but the quantities measured are generally variable ones, themselves subject to chance. Under such circumstances, a single determination proves nothing and indicates little. On the other hand a series of determinations not only reduces the probable error of the result, but furnishes intrinsic evidence of its value. Consider, for example, a series of four measurements made upon a single constant quantity by an observer, A. These are 103, 98, 101, 98, the average of which is 100. Assuming 100 to be the correct result (it is the best obtainable consistent with the observations, although, of course, not correct) then the error of each observation is the difference between it and the mean, or +3, −2, +1, −2, the plus and minus signs indicating whether the errors make the individual determinations greater or less than the mean. Now, if another observer measures the same quantity with the results 101, 100, 102, 101, averaging 101, the errors of his determinations will be 0, −1, +1, 0. It would be concluded at once that B's measurements and result of 101 were more accurate than A's because his errors, figured from the mean, are less than A's.

Compound Quantities. Most engineering measurements are upon compound quantities, the components of which are variable. The following parallel illustrates this case and also the graphic method later to be described. Fig. 1 represents the floor of a room; and the solid diagonal line, the path of a ball that rolls across it. It is desired to locate as exactly as may be the path as it is traversed by the ball. The only exact data we have are that it is a straight line and starts at 0. At a given instant, one observer measures the distance of the ball from the wall OX and another from OY. These distances, x and y, are sufficient to locate one of the positions of the ball and therefore its path, provided they are accurately determined. Accidental errors prevent this, however, so that the path is falsely located on the line Oa. Obviously, if a number of points, instead of only one, could be located, a better result would ensue. b, c, and d are such points. It is then found that these points do not lie on the

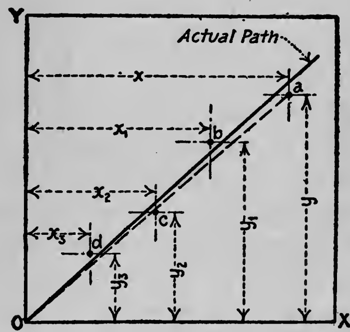


FIG. 1.

same straight line, the distances from the true path being the errors of their determinations. The question now arises how should these discordant observations be made to agree upon one result, which, though perhaps not the true one, is in best accord with the data. It is possible to locate a different line with each pair of observations, $x \div y$ locating one, $x_1 \div y_1$, another, and so on. One might conclude that the average value of $x \div y$ would be best, but it can be shown by the theory of errors that the arithmetical mean does not yield the best result upon indirectly determined quantities. In this case, the result is obtained by measuring two other quantities; the determination

is therefore indirect. The best result (referred to generally as "most probable") is a line so located that the sum of the squares of the distances from it of the points *a*, *b*, *c*, and *d* shall be a minimum. Such a line can be determined mathematically, but it is generally drawn by eye judgment.

Variables, Independent and Dependent. A large part of mechanical experimentation is parallel to this simple illustration. Two variables, bound together by a more or less rigid law, are measured, and from the result the law is deduced. The law is not always represented by a straight line, but often by a curve. Whichever it is, it is the business of experimentation to find.

One of the variables can always be controlled; the other then follows it according to the law binding them. For example, when a spring is extended by a force, there are two quantities, namely, the force and the extension, which combined give the law of the spring. In their measurement we may add predetermined increments of weight and let the extension vary as it may, or we may add weight enough to cause predetermined increments of extension and let the weight increase as it may. The predetermined quantity is called the "independent variable"; the other, "dependent."

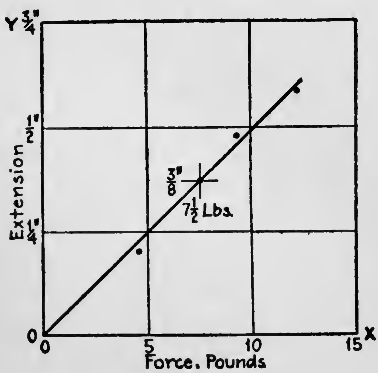


FIG. 2.

It is almost always desirable to present such measurements as points and to locate by them a smooth curve. This is done by adopting a pair of rectangular axes, *OX* and *OY* and scaling them according to the units measured. In Fig. 2, for instance, 1 in. along *OX* represents 10 lbs. applied to a spring and 1 in. along *OY* represents ½ in. of

spring extension. The observations 7.5 lbs. and 0.375 in. are thus shown by the point indicated, and so on.

In many cases there are three variables, as the pressure, temperature and volume of a gas. It is then customary to keep one of them constant throughout a test, arbitrarily control another, constituting the independent variable, and let the third vary as it may.

If the relation between the variables appears to be satisfied by some other curve than a straight line, such a one is drawn.

Intrinsic Evidence of Accuracy. If any point of a plotted series lies markedly off the line, it may be concluded logically that its determination contained some large error, or mistake, and it may be thrown out of consideration entirely. Similarly, the value of a set of measurements may be judged by the closeness of the plotted results to a smooth curve.

It will be readily seen that the value of graphic presentation of results lies in the fact that their relative variation is visualized and can therefore be grasped by the mind much more easily than could mere columns of figures.

Conventions for Plotting. The independent variable, as a matter of convention, is plotted horizontally, the other vertically. Exceptions to this rule are calibration curves and stress-strain diagrams of materials; the one having instrument readings, the other, deformations, plotted as abscissas.

Compound Variable Quantities. In the case of the spring the two variables, when combined, give a constant quantity, namely, the number of pounds per inch of extension. It is not often that the required compound quantity is a constant one, although it is sought to keep it so throughout one set of measurements. As an example, the mechanical efficiency of a steam engine depends upon the amount of work it is doing; the greater its output, within its capacity, the greater its efficiency. Suppose it is desired to measure the efficiency at a given output. The engine is operated to deliver this horse-power, no more and

no less. But, owing to variations in steam pressure, governor action, and the like, it is possible to keep the horse-power output only approximately constant; it fluctuates somewhat through accidental causes more or less outside our control. Further, even if it were possible to keep it constant, the internal friction of the engine varies and this causes the work done in the cylinder to vary. Consequently, the efficiency, which is the delivered horse-power divided by the power developed in the cylinder, will vary. So aside from variations in the results produced by unavoidable errors of measurement, there are accidental variations in the true value of the result, even when all the controllable conditions are kept as constant as possible. It is like measuring the height of a small spot of light which persists in fluctuating up and down through a limited space. Clearly, a single measurement would be altogether insufficient, for it might represent a high or a low value, even if accurately made. Obviously, the best that can be done is to keep controllable conditions as constant as possible while taking not one but a number of measurements. It may now be understood how such conditions will determine the precision necessary, because of the uncertainty of the true value of the result.

Rules for Testing. The following rules should be observed when measuring variable compound quantities.

First. For one value of the independent variable, the series of measurements made of each separate quantity may be averaged. All such averages may be used in a single calculation of the desired result.

Second. Observations of fluctuating quantities should be made at equal time intervals to secure a true average.

Third. The number of observations necessary of any quantity depends upon the constancy of the quantity.

Fourth. Tests of time rates should cover a sufficient duration of time to reduce the error of initial and final measurements to 1 per cent of the total quantity.

Fifth. In time rate tests, intermediate time readings should be taken to check rate constancy.

Constancy of Conditions. Referring to the third rule, if all the conditions of a test could be kept absolutely constant, only two sets of readings would be necessary. The two would give identical results; the second would be made only to check the accuracy of measurement of the first. In some cases this condition is approximated, as, for instance, electric motor driven blowers. In other cases some of the quantities fluctuate more than the rest; under these circumstances the time interval between measurements of these quantities should be smaller than for the others, so that there may be more values obtained for them.

Duration of Time Rate Tests. The fourth rule may be illustrated as follows. In measuring the pounds of water per minute flowing into a tank, the cross-section of the tank is measured, and the difference of water in it before and after an observed time interval is figured by noting the corresponding difference of the water levels. The water level is measured with a scale graduated to tenths of an inch. With such a scale, it is possible to make an error of $\frac{1}{20}$ in. (half a division) at each measurement, totaling $\frac{1}{10}$ in. It would be necessary, then, to make the duration of test long enough that $\frac{1}{10}$ in. be 1 per cent of the total rise in water level, that is, long enough for the water to rise 0.1 in. $\div$ 1.0 per cent $=$ 10 ins. This calculation allows for the maximum error probable, but it is on the safe side.

Intermediate Readings. The table of observations on p. 10 illustrates the fifth rule. If the column of differences shows approximately equal values corresponding to equal time intervals, the result is valid. Any marked departure from constancy should throw suspicion upon the result. If the readings at 3:00 and 3:10 only had been taken, we should be ignorant as to rate constancy. As has been pointed out, the reliability of the result depends upon constancy of conditions.

Time	Height	Differences
3:00	5.0 in	
		0.95 in.
3:02	5.95	
		1.05
3:04	7.00	
		1.05
3:06	8.05	
		1.00
3:08	9.05	
		1.05
3:10	10.10	

A further advantage from intermediate readings is that they strengthen the determination against mistakes. This was shown in the illustration of the path of the ball. From the observations just cited, the result ordinarily would be figured thus

$$\frac{(10.1-5.0)\times \text{area}}{10 \text{ min.}} = \text{cubic inches per min.}$$

which is the same as though no intermediate readings had been taken. If, however, a mistake in measurement were incurred in the first or the last reading, as 6.0 instead of 5.0 ins., or 20.1 instead of 10.1 ins., this mistake would not be apparent, and the result misleading or worthless. Intermediate readings would disclose such a mistake, and the set of observations involved could be discarded without sacrificing the others.

In time quantity measurements it is always best to record the time of day instead of time intervals merely. This insures a correct record of the time. Otherwise it is easy to note an interval such as three minutes when two or four actually have elapsed. A further advantage is that possible irregularities in results may then be accounted for by related happenings that might be associated only by a knowledge of the time of occurrence.

Before closing the general subject, it is well to emphasize the importance of figuring in the laboratory rough results from the observations as soon as obtained. This applies to commercial as well as student work. By so doing, many faults in operation and in the application of instruments may be detected and remedied in time. It is especially valuable to plot a curve of the variables as the test progresses, for it will reveal the accuracy or error of the determinations at the time when such knowledge is most valuable.

Problem I_1. Temperatures of liquid in a pipe are read as follows: At 11:00 A.M., 90°; at 11:05, 100°; 11:15, 100°; 11:20, 90°. Compare the average of these four readings with the probable average if a reading had been taken at 11:10. *Ans.*, 95°, 96°.

Problem I_2. If the thermometer used in Problem 1_1 has a least count of 2°, what is the probable percentage of error in each reading?

Ans., 1.1%, 1%.

Problem I_3. In a specific heat determination, two thermometers are used reading about 50° and 70°, respectively. What should be their least count that the probable error of the result be less than 2 per cent?

Ans., 1°, or less.

Problem I_4. The rate of water flowing into a tank placed on a platform scales is to be determined by taking weighings and timing. If the rate is about 20 lbs. per minute and the probable error in reading the scales at starting and stopping is $\pm\frac{1}{2}$ lb., how long should the test be continued to secure less than 1 per cent of error? How many intermediate readings could be conveniently taken in this time? *Ans.*, about 5 min.

PART ONE

THE TESTING OF INSTRUMENTS

WEIGHTS AND FORCES

THESE are measured by comparison with known weights with the aid of a system of levers such as in a beam balance, or by reference to the amount they deform some elastic object, as a spring, which has been previously calibrated against standard weights. When a leverage system is used so that a large force may be measured by comparison with a small known weight, it is necessary to know the ratio of the lever arms. In Fig. 3, for instance, the force F is measured by the relation

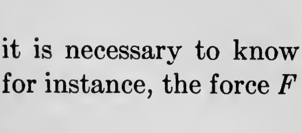


FIG. 3.

$$W = \frac{n}{m} F$$

in which $\frac{n}{m}$ may be referred to as the "leverage ratio."

The calibration of any force measuring appratus rests primarily upon the force of gravity as a standard. "Standard weight" is a misnomer in that the *weight* of the body so called, being the

force of gravity between it and the earth, actually varies with the locality of the body. Standard mass is a better term. A standard mass establishes a standard force when the acceleration of gravity at the given locality is known.

1. Calibration of Platform Scales

Principles. The platform scales Fig. 4, is arranged so that a large weight, W to be measured, may be balanced by a small

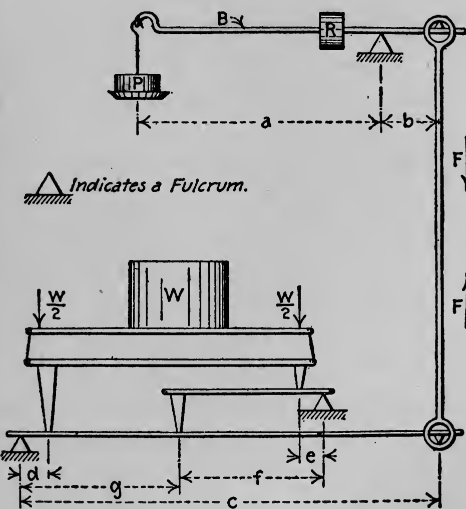


Fig. 4.—Platform Scales.

weight on the beam, B. The small weight is either or both P on the poise or R, the rider, acting at a variable distance from the beam fulcrum. If P balances W, then by the principle of moments,

$$P\times\frac{a}{b}=F=\frac{W}{2}\times\frac{d}{c}+\frac{W}{2}\times\frac{e}{f}\times\frac{g}{c}$$

It is arranged that $\frac{d}{c}=\frac{e}{f}\times\frac{g}{c}$ so that

$$P\times\frac{a}{b}=W\times\frac{d}{c}$$

and

$$\frac{W}{P}=\frac{ac}{bd}=\text{the leverage ratio.}$$

(a) **The Sensitiveness of the Scales.** This is determined by placing a large weight on the platform, weighing it, and then finding the smallest additional weight that will cause a deflection of the beam which can be nicely balanced by the rider. This additional weight is also a measure of the precision of the scales, provided the beam graduations are small enough to take cognizance of it.

(b) **Beam Calibration.** Readings of the beam are taken corresponding to a number of standard weights. These should be sufficient to cover fairly the range of the beam. If standard weights of convenient size are not available, a number of small weights may be standardized for the purpose by using a calibrated scales with slightly greater precision than that of the scales to be tested. (See p. 3.)

If the instrument readings do not agree with the true weights a calibration curve should be plotted, having the instrument readings as abscissas and the true weights as ordinates. This curve may be used to get corrected values at any part of the beam.

(c) **The Leverage Ratio** may be found by measurement of the lever arms, but this method is difficult and subject to error. A better one consists of balancing a standard weight on the poise with a standard weight on the platform, then calculating the ratio of these weights. The nominal leverage ratio may be learned by examination of the poise weights. They are marked with their actual weights and the weights they are intended to balance, as, for instance, 1 lb.–100 lbs. This gives a nominal leverage

ratio of 100. To test the accuracy of this, place, for example, a standard $\frac{1}{2}$-lb. weight on the poise and a standard 50-lb. weight on the platform. If they do not balance, add enough weight (shot is convenient) to either one or the other until a balance is secured. This additional weight may then be accurately measured and the true leverage ratio found.

(d) Poise Weight Calibration. The indications of the poise weights are accurate if the leverage ratio is true and if the actual weights of the poise weights are as marked. They should therefore be weighed by a calibrated scales of sufficient precision. It should be noted that any error in the poise weight is multiplied in the instrument reading by the leverage ratio. The standard scales should therefore weigh these weights to within an amount equal to the precision of the scales to be tested divided by its leverage ratio.

If the actual weight of a poise weight is not as marked, or if the leverage ratio is not true, then the weight balanced on the platform is

$$\text{Actual weight of poise weight} \times \text{actual leverage ratio}$$

instead of the amount indicated by the marking. For instance, if the poise weight intended to measure 100 lbs. actually weighs 0.99 lb. instead of 1 lb., and if the leverage ratio is 99.5 instead of 100, then the weight on the platform balanced by it is $0.99 \times 99.5 = 98.5$ instead of 100 lbs. In this way the true weight balanced by each poise weight may be found.

(e) Beam Calibration by Test of Rider. In some types of platform scales, there are no poise weights, a number of riders on different beams being used. In such cases, the following method may be used:

The weighing beam to be tested is represented by Fig. 5. With no load, the rider is in the position shown, its pointer being at the zero graduation, and its beam floating. With a load W on the table, the rider must be moved through a distance d to

secure balance. If the scales are correct, the weight of the rider must be such that the distance d will be that between the zero and the graduation marked W. Now, instead of using the rider, another weight could be applied at any convenient part of the beam as shown by S, of such amount as to secure a balance when the load W is on the table. Therefore, the moment of the weight S about the fulcrum f must equal the moment of

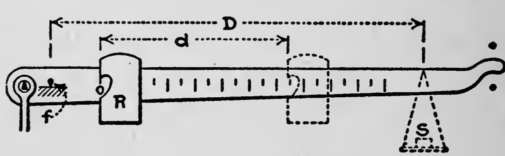


FIG. 5.

the rider which is replaced. Letting R denote the weight of the rider,

$$S \times D = R \times d$$

Let the ratio of the load on the platform to the balancing weight S be L. Then

$$S = \frac{W}{L}.$$

Substituting this in the first equation, and simplifying, we find the weight which the rider must have in order to suit the existing graduation of the beam:

$$R = \frac{D}{d} \times \frac{W}{L}.$$

To apply this relation to the calibration, a value of W is chosen, and the distance between the beam graduation marked W and the zero graduation is carefully measured. This determines W and d. A point is then chosen to represent D, and its distance measured from the fulcrum f. To find the ratio L, it is not necessary to apply the full weight W to the platform,

since the ratio $W : S$ is constant for all values of W, the ratio of levers being established by the selection of D. Therefore the following procedure is used. An appreciable, but not inconvenient, number of weights is placed on the platform, these having been previously measured with a standard scales. They may consist of anything available that may be readily moved. A balance pan, improvised from paste-board and wire, is then attached to the beam at the distance D from the fulcrum, and to this pan are added shot or other small weights until a balance is secured. If desired, the weight of the pan may be previously balanced by the usual beam counterweight so that the experimenter need deal with the added weight only. The latter should then be accurately weighed. This result, divided into the weight applied to the table, gives the desired ratio L. The value thus obtained completes the data necessary to calculate the weight of the rider according to the equation previously deduced. The rider is then removed from the beam and weighed; if the actual checks the calculated weight, the instrument is proved at the graduation marked W. Any other graduation may then be checked by proportion, since the distances of the graduations from the zero mark must vary directly with the indicated loads.

Problem 1_1. Five weights of about 10 lbs. each are to be standardized in order to calibrate a beam up to 50 lbs. How closely should these secondary standards be weighed, if the least count of the beam to be calibrated is 4 oz.? (Note. Have regard for the cumulative error.)

Problem 1_2. Referring to division (c), how closely should the shot be weighed for less than 1 per cent of error in the determination of the leverage ratio?

Problem 1_3. The least count of a scales to be calibrated is 4 oz. Its leverage ratio is 100 : 1. How closely should the poise weights be weighed so that their calibration shall be within the precision of the beam? See (d).

Problem 1_4. Prove that it makes no difference in the instrument indications where W is placed on the platform.

Problem 1_5. If the rider, R, is too light, will the resulting error be constant at all indications of the beam or will it vary and why? Will the error be plus or minus?

Problem 1_6. If the poise is too light to bring the beam down with nothing

on the platform, can a scales, otherwise accurate, be used for correct results without previous calibration, and how? If too heavy?

Problem 1_7. Calculate the proper weight of the rider and check by weighing it.

Problem 1_8. Examine a scales to find if its leverage ratio may be adjusted.

2. Evaluation of a Spring

Principles. In a large class of instruments, it is necessary to know the amount of force required to extend, compress, or twist a spring per unit of deformation. This quantity is called the "spring scale." According to Hooke's law, it is a constant within the elastic limit of the material. In many cases the movable end of the spring is attached to some combination of links and gears designed to indicate, magnify, or translate the motion. When the applied force is increasing, the friction of such links or gears makes the instrument read low, since the indicator is moving up the scale and friction tends to hold it back. With a decreasing force, the instrument reads high for a similar reason. Suppose an external force F is applied to a spring instrument, which is correct except for the effect of friction, first by increasing the external force to the value F, and then by decreasing it to this value. Then, if S is the spring scale, E_1 and E_2, the extensions in the two cases, and X the friction,

$$F = S \times E_1 + X$$

Adding

$$\begin{aligned} F &= S \times E_2 - X \\ \hline 2F &= S(E_1 + E_2) \\ F &= \frac{S(E_1 + E_2)}{2} \end{aligned}$$

That is, the effect of friction may be eliminated by taking increasing and decreasing readings at each load applied, dividing the sum of the extensions thus found by two, and using the result as the extension caused by the external force.

(a) Spring Scale by Graphic Method. Take as an example the following measurements of force and extension.

Force lbs.	Extensions		
	Increasing In.	Decreasing In.	Average In.
10	0.32	0.34	0.33
20	0.64	0.70	0.67
30	1.02	1.08	1.05
40	1.32	1.36	1.34
50	1.68	1.70	1.69

These are plotted as points on coordinating paper, and then is drawn what appears to the eye as the best straight line to satisfy each of the three sets of points.* Fig. 6 shows the line for the increasing readings; the other two curves are similar. When it is desired to apply a calibration only to increasing readings, the ascending line is used, and similarly with the descending. The combined line gives the best results when the instrument is used for measurement of a fluctuating quantity.

To figure one of the spring scales, as for instance the ascending, the line for which is shown by Fig. 6, a point is selected near its extremity, as $F_1 - E_1$, this point not necessarily representing a pair of observations. It should, however, lie *on* the line. Then the spring scale equals

$$S\text{ (ascending)} = \frac{F_1 - k}{E_1}$$

or, using numerical values,

$$= \frac{47.4 - 1.0}{1.60} = 29.0 \text{ lbs. per inch}$$

* Certain researches of the author indicate that the decreasing values of force and extension do not follow Hooke's law within appreciable amounts and that, theoretically, they must follow a curved line. (Physical Review, October, 1911.) To represent them by a straight line is therefore merely a convenient approximation.

to be used when the readings of the instrument increase; similarly with the other two spring scales.

Note that the intercept, k, is taken into account. If the curve does not pass through the origin it is because of friction or backlash of the indicating mechanism or to a false zero reading of the extension.

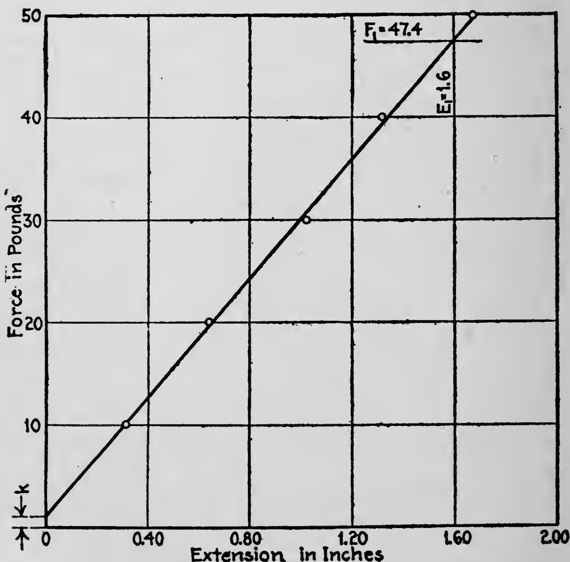


Fig. 6.

(b) Spring Scale by Method of Least Squares. The theory of errors shows that the most probable straight line to fit such data is one so located that the sum of the squares of the distances of the plotted points from the line shall be a minimum. This can be calculated as follows. The equation of the line may be written

$$F = SE + k$$

in which k and S are unknown, and F and E are represented by a number of more or less discordant observations. This equation may be multiplied through by the coefficient of each unknown, thus

$$F = SE + k$$
$$EF = SE^2 + Ek.$$

In each equation the corresponding values of E and F are substituted thus forming two series of equations. Each series is summed and the resulting equations are known as the "normal equations" of k and S respectively. These may be solved as simultaneous equations to obtain the most probable value of S.

Using the data of the ascending scale tabulated,

$F = ES + k$	$EF = E^2S + Ek$
$10 = 0.32S + k$	$3.2 = 0.1024S + 0.32k$
$20 = 0.64S + k$	$12.8 = 0.4096S + 0.64k$
$30 = 1.02S + k$	$30.6 = 1.0404S + 1.02k$
$40 = 1.32S + k$	$52.8 = 1.7424S + 1.32k$
$50 = 1.68S + k$	$84.0 = 2.8224S + 1.68k$
$150 = 4.98S + 5k$	$183.4 = 6.1172S + 4.98k$
Normal equation of k.	Normal equation of S.

From these, by eliminating k, it is found that $S = 29.38$ lbs. per inch.

The descending scale is found similarly and the combined scale is obtained by adding the corresponding normal equations from the ascending and descending values.

Note that the numerical values in the equations should be figured to four or five significant figures since, when the equations are solved, the first two or three figures disappear by subtraction. Labor can be saved by using multiples of ten, or simple figures, for the independent variable F, and by using a pocket-book table of squares to obtain the values of E^2.

Problem 2_1. Calculate the descending and combined scales for the example given by methods (*a*) and (*b*). Compare them.

Problem 2_2. Prove from the fact that the area under each curve equals work done that the descending curve cannot be a straight line.

Problem 2_3. From the ascending data of the example given, figure and compare the different results for the ascending spring scale obtained by the following (faulty) methods. $50 \div 1.68$. The average of all the F's $\div$ average of E's. The average of $F \div E$. The average of each increment of $F \div$ corresponding increment of extension. The average increment of F divided by average increment of E.

PRESSURE

The measurement of pressure is a special case of force measurement, wherein the area over which the force is distributed is taken into account. Pressure is a compound unit being the number of units of force per unit area.

In calibrating pressure measuring devices, the standard force is that of gravity acting on some standard mass.

Pressure is usually expressed in pounds per square inch. Its measurement is always relative, that is, based upon some other pressure as a datum. A pressure gage reads so many units *above* atmospheric pressure, for instance, and a vacuum gage, so many *below*. Absolute pressure, or pressure counted from zero, is an abstract conception and cannot be measured directly by instruments.

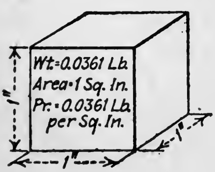


Fig. 7.

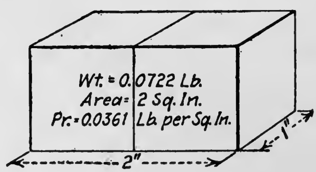


Fig. 8.

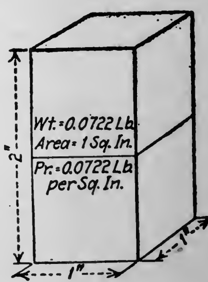


Fig. 9.

Pressure Produced by Water.

Another measure of pressure is the height of water or other liquid which by its weight balances the pressure to be measured.

Fig. 7 represents a cubic inch of water. At 60° F., this weighs 0.0361 lb. Since the area this force acts upon is one square inch, the pressure produced is 0.0361 lb. per square inch. Two cubic inches of water, arranged as in Fig. 8, would have twice the weight, but it would be imposed upon twice the area; hence the pressure would be the same. Arranged as in Fig. 9, however, the area would be one square inch, and therefore the pressure would be twice that shown by Fig. 7. It follows that the pressure produced by a given mass of water varies directly with its height and is independent of its cross-section. Hence, inches or feet of water may be regarded as units of pressure.

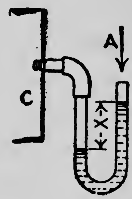


FIG. 10.
Manometer.

Fig. 10 shows the application of this principle in the "manometer." The pressure in the chamber C is said to be "X inches of water" and this equals $0.0361 \times X$ lbs. per square inch. The absolute pressure in C equals this quantity plus the pressure of the atmosphere represented by the arrow at A.

Mercury is also used in manometers, and since its specific gravity is about 13.6 at 60° F., the equivalent is

$$1 \text{ in. of mercury} = 0.0361 \times 13.6 = 0.49 \text{ lbs. per square inch.}$$

3. Calibration of a Bourdon Gage

Principles. The Bourdon gage consists essentially of a hollow circular spring which is deformed when subjected to internal fluid pressure, the deformation causing a pointer to rotate upon a graduated dial. (See Fig. 11.) The pointer is readily removed so that it can be set at any part of the dial to correspond to an applied known pressure.

It should be noted that the deformation of the hollow tube is proportional not to the absolute pressure within the tube, but to the difference of pressure within and without. The Bourdon pressure gage, therefore, always indicates pressures above at-

mosphere, since the outer surface of the tube is subjected to atmospheric pressure; and to convert its readings into absolute pressures, one must add the barometric pressure expressed in the same units.

The vacuum gage works on the same principle as the one just described, the only difference being that the excess of pressure is on the outside of the tube (Fig. 11) causing a contraction of the coil, instead of an expansion, which is indicated, generally, in inches of mercury less than the barometric.

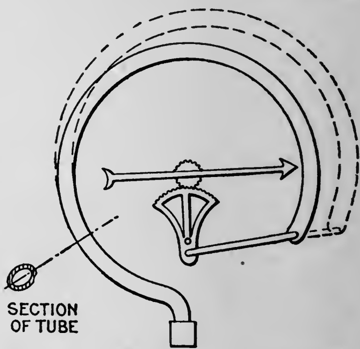


FIG. 11.—Mechanism of Bourdon Gage.

The testing apparatus consists of a chamber in which any desired pressure may be obtained and to which is attached some accurate device for measuring it. The pressure is generally obtained by some simple form of hand pump. The measuring device is either a "test gage," a column of mercury, or a set of weights acting on a plunger of known area arranged to produce the pressure desired. The test gage is not a desirable standard as it needs calibrating itself from time to time. The mercury column is a cumbersome and expensive apparatus for pressures above a few pounds, although it is the most accurate method of measuring the true pressure. Besides, a high degree of accuracy is not needed since the least count of commercial gages is generally not less than 5 lbs. It is necessary, however, to use the mercury column when testing vacuum gages, and convenient since they are graduated in inches of mercury and not in pounds per square inch. (See Fig. 12.) The standard weight

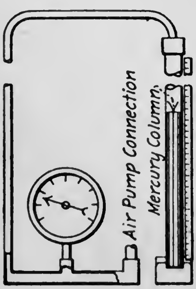


FIG. 12.—Vacuum Gage Tester.

method is convenient but the friction of the plunger prevents true records. Fig. 13 shows an apparatus of this type.

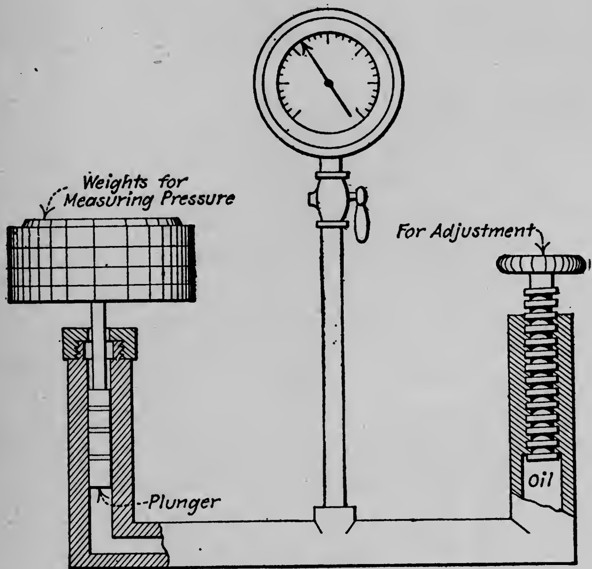


Fig. 13.—Dead Weight Gage Tester.

(a) **The Constants of the Standard Weight Apparatus** are the area of the plunger and the actual weights of the test weights, from which may be figured the actual pressures produced by them. The area is measured by calipering the plunger. The weights should be determined by comparison with standard weights with a degree of precision compatible with the least count of the gage to be tested. Note that the pressure produced by the plunger and attached pan is always applied.

(b) **Calibration.** Set the pointer to read accurately at the division most used and then take a series of readings of true pressures and instrument readings increasing and decreasing, from which plot a calibration curve. Convenient procedure

is as follows. With the weights applied to produce the desired pressure, depress the plunger a trifle by hand and close the cock between the gage and the pressure chamber, thus confining in the gage a pressure slightly greater than produced by the weights. The pan and weights are now revolved to reduce friction at the plunger and the cock slowly opened. The pressure indicated by the gage will then slowly decrease and a reading may be taken. For increasing values, the pan is raised a trifle, otherwise the procedure is the same. The two readings thus found are added and divided by two to eliminate the effect of friction. If the gage is to be used for ascending pressures, then the calibration resulting from them only should be used. Generally this is not the case and the calibration from the combined readings is preferable.

(c) **Adjustment of the Scale.** When the calibration curve is plotted, if the instrument is in error, it will be noted that the indications either increase or decrease with relation to their true value. Inside of the gage will be found an adjustable link, by means of which the travel of the pointer may be made greater or smaller for a given motion of the tube. This link should be changed in length until a correct motion of the pointer on the scale is found, as proved by a calibration curve.

(d) **The calibration of a vacuum gage** is essentially the same as for a pressure gage, an air pump and mercury column being used instead of the apparatus described.

(e) **Calibration of a Recording Pressure Gage.** The working principle of the usual pressure recorders is that of the Bourdon tube, the tube being helical in form instead of circular. This provides a sufficient motion to the free end of the tube, to which a pen arm is attached; and magnification by links is not needed. The pen arm swings over a circular chart which is uniformly rotated by clockwork. The curve traced by the pen is thus one of pressure shown radially against time circumferentially. (See Fig. 14.)

When calibrating, the clock should be stopped, and the instrument read the same as a simple indicating gage. The deadweight apparatus may be used, the pen arm first being set to indicate accurately at the desired point on the chart scale.

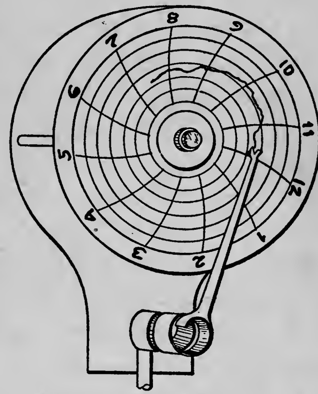


Fig. 14.—Mechanism of Pressure Recorder.

Recorders are often set at a considerable distance from the points at which the pressure is to be ascertained. If the gage is either above or below such a point, and the connecting tube is full of liquid (such as condensed steam when the gage is below a steam pipe whose pressure is sought) then a correction for the head of liquid must be applied. Since 2.3 feet head is equivalent to 1 pound per square inch, the correction for a difference of height of H feet is

$$H/2.3 \text{ lbs. per square inch,}$$

and this should be subtracted from the instrument indications when the gage is set below the point of measured pressure, and added when above.

Problem 3_1. If the table and plunger of the test apparatus weigh $15\frac{1}{2}$ oz., how much pressure does their weight produce in pounds per square inch, the diameter of the plunger being 0.50 inch? Is the difference between this and 5 lbs. per square inch worth considering?

Problem 3_2. If the area of the plunger of the test apparatus is about $\frac{1}{8}$ square inch, how closely should the test weights be weighed to come within the precision of a gage having a least count of 5 lbs.?

4. Draft Gage Calibration

Principles. Draft gages are generally special forms of manometers used to measure the reduced pressure (less than atmosphere) in chimneys, which creates the draft. This is expressed in inches of water and, as it is usually only a few tenths, the ordinary

U-tube will not do. Fig. 15 shows a favored type. It is the ordinary manometer with one leg on a slant, so that the travel of the level of the liquid used is magnified. In use, the instrument should be set true to its designed level and the liquid adjusted so that the level in the inclined tube stands at zero when there is no difference of pressure.

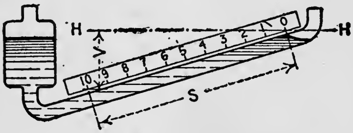


FIG. 15.—Draft Gage.

Calibration of a draft gage can be made by examination of its graduations and the weight of the liquid, or by comparison with a standard instrument.

(a) **Calibration by Calculation.** Having set the gage to its designed level, a horizontal line H–H, is drawn with a spirit-level through the zero of the scale. The vertical distance V from the last graduation on the scale to this line is then measured. The level of the liquid in the inclined tube will fall this distance for the whole scale, but the level in the enlarged tube will rise an amount equal to the length S of the scale in inches multiplied by the ratio of the squares of the diameters of the bores of the small to the large tube. If water is the liquid, the sum of the rise and fall thus calculated equals the inches of water, pressure. The true pressure corresponding to any other graduation can now be found by proportion.

The calculation of the rise of level in the enlarged tube assumes that the bores are uniform.

Commercial forms of this type of gage generally use oil for the liquid, having a definite specific gravity less than one. When such a gage is calibrated, the total vertical difference of level is found as before. The corresponding height of water is then found by dividing this by the specific gravity of the particular liquid used. The specific gravity may be obtained by using a hydrometer or by weighing a known volume of the liquid.

(b) **Calibration against a Standard Gage.** Connect the gage to be tested with a piece of rubber tubing to the standard

gage as in Fig. 16. The pressure in this tube may readily be reduced to any desired amount by applying the lips to the branch B, the pinch-cock serving to confine the suction when obtained. Enough readings for a calibration curve should be taken.

For the standard gage, an impromptu instrument is easily made by using an inclined tube of generous length.

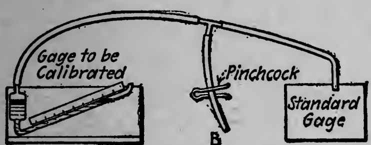


Fig. 16.—Arrangement for Testing.

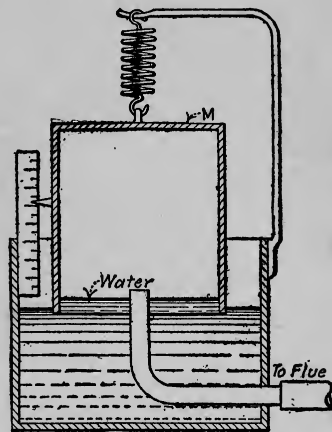


Fig. 17.—Kent's Draft Gage.

Problem 4_1. If a draft gage using water at 70° F. is correct, calculate the rise in temperature that would produce 1 per cent of error. (See p. for weight of water.) *Ans.* 60°.

Problem 4_2. In the type of gage shown by Fig. 17, the reduced pressure inside the inverted can M causes it to descend against the resistance of the spring (neglecting the buoyancy of the water). If the area of the inverted can is 50 square inches and if the pressure within is $\frac{1}{10}$ inch of water less than without, what is the total downward force on the can? How much will the spring extend because of this force if the spring scale is 0.2 lb. per inch? *Ans.*, 0.9 in.

Problem 4_3. Deduce an equation for the gage shown by Fig. 17, neglecting buoyance, to show the relation between the extension of the spring and the draft, in inches of water, causing it.

ANGULAR VELOCITY

The units are usually whole revolutions per minute, abbreviated R.p.m. The simplest form of instrument to measure this quantity is the **hand counter.** This consists of a worm and wheel. The worm is part of a small spindle tipped with rubber so that,

if held against the end of a revolving shaft, it turns with the shaft, thus giving a circular motion to the worm wheel. The wheel is graduated so as to indicate the number of turns of the spindle. In operation, the hand counter is applied during a period of time measured with a watch, and from the readings of watch and counter, the revolutions per minute of the shaft are calculated.

The Continuous Counter, or cyclometer, is a variation of the hand counter in that its spindle is geared to a series of wheels with numbered faces, partially exposed, so as to show at any time the total number of revolutions. Some forms of continuous counter are driven by a reciprocating lever instead of a revolving spindle. In both forms, the instrument usually receives its motion from a small pin on the end of the shaft whose revolutions are to be measured, which pin acts as a crank.

The Tachometer gives instantaneous indications of revolutions per minute without measuring time. The spindle transmitting the speed bears a pair of weights so linked that they move outward from the spindle by centrifugal force, and against the restraint of a spring. In so doing they actuate a pointer on an appropriately graduated dial. As the centrifugal force, and therefore the motion of the weights. are proportional to the speed, the pointer may register the speed instantaneously.

On account of the variation in their internal friction and in the stiffness of their springs with use, tachometers should be calibrated before using on important work.

Recording Tachometers are made in a variety of forms, and, for the greater part, depend upon the action of centrifugal force on a solid or fluid mass. Thus, the simple tachometer just described may be made as a recorder if the centrifugal weights actuate a pen arm (instead of a dial pointer) which travels over a clock-work driven chart. An example of the centrifugal fluid tachometer is the Bristol. This consists of a small air blower which is driven by the shaft whose speed is to be found. The blower creates a partial vacuum which increases as the speed

increases, and vice versa. The vacuum is transmitted to and recorded by a pressure recorder, the chart of which is graduated in revolutions per minute. Still another recording tachometer consists of a small generator, driven by the shaft considered, whose speed variations are evidenced by the generator voltage. This is recorded in terms of revolutions per minute upon a time chart.

The Chronograph is an instrument by which a graphic record of time is made. Fig. 18 shows a form. Paper is caused to roll over a drum, D, by clockwork, and in contact with the paper is a pen on a light arm, A, which is caused to vibrate at regular time intervals by connection, electrical or otherwise, with a standard clock. Another arm, A', is similarly actuated by connection with the shaft the speed of which is to be measured, so that it vibrates once each revolution. Thus, there are drawn two lines, one broken at intervals representing time, the other, revolutions; from which may be obtained angular velocity.

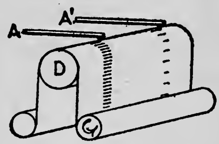


FIG. 18.—Chronograph.

The Tachagraph is a very sensitive tachometer, arranged to give an autographic diagram of angular velocity. It is used to measure minute changes of speed within a revolution.

5. Calibration of a Tachometer

Principles. The standard should be either a continuous counter or a chronograph. In either case it is necessary to have a revolving shaft of variable and controllable speed to cover the whole range of operation of the tachometer. A small steam engine of variable speed may be used, or a water wheel, or a series wound electric motor. A shunt wound motor will do if a rheostat is put in series with its armature circuit. A small water rheostat is convenient as it giver a wide range of resistance.

(a) Calibration against a Continuous Counter. Increasing and decreasing readings of the tachometer should be taken at

each speed. The mean of these readings is plotted against the speed as shown by the counter for the calibration curve.

If the continuous counter is arranged on the variable speed shaft so that it can be applied at the same time as the tachometer, the procedure is as follows. Starting with a slightly lower speed than is required, the tachometer is connected and the speed raised to the desired amount. Time and counter readings are then taken with the tachometer still connected. If the speed remains constant as indicated by the tachometer, the observations are valid after the second readings of time and counter are taken. The decreasing readings are obtained similarly.

When the speed is high the counter becomes difficult to read. This may be obviated by an arrangement for throwing the counter in or out of connection with the variable speed shaft.

A hand counter may be used if it can be applied to one end of the shaft while the tachometer is at the other. It is inadvisable to take readings of the two instruments at other than the same time as the speed may vary.

The time during which the revolutions are counted should be of sufficient length to reduce the error of starting and stopping to less than the precision of the tachometer. For instance, if the tachometer cannot be read closer than 5 R.p.m., at any part of its scale, then at 250 R.p.m. the error in reading the instrument is 2 per cent, and the error in reading the time should be made less than this, say 1 per cent. Now a moving number is unlikely to be timed with an ordinary watch closer than one-half second; therefore the counting should proceed through at least 50 seconds to get the desired precision. Similarly, at 500 R.p.m., the error in reading the instrument is 1 per cent; the starting and stopping error should be reduced to $\frac{1}{2}$ per cent, and the timing should proceed through 100 seconds. The use of a stop-watch will greatly reduce this time, as it reduces the starting and stopping error to one-tenth of a second or less.

(b) Calibration against a Chronograph. With this apparatus the tachometer may be left in connection with the variable speed

shaft, and the speed first increased and then decreased in a series of steps at constant speed. The moving record may be stopped while the speed is being adjusted. The time for counting may be very much reduced, as the chronograph shows the whole number and fraction of turns of the shaft that occur during a time beat which may be as small as desired.

(c) Calibration of a Recording Tachometer. The principles to be observed are exactly the same as given under (*a*) and (*b*). It should be observed that certain types of this instrument have a time lag in their indications behind the true R.p.m. of shaft whose speed it is intended to measure. That is, when this shaft changes in speed, a certain interval is required before the change is felt at the recorder. This interval should be noted and reported.

Problem 5_1. Using an ordinary watch, how long should a continuous counter be timed when calibrating a tachometer witl a range from 600 to 1200 R.p.m., at the even hundreds, if the least count is 10 R.p.m.? How long, if a stop-watch is used?

POWER

Power is defined as the time rate of work. Quantitatively, work is the product of a force and the distance through which it acts, the unit being foot-pounds. A horse-power is defined as 33,000 foot-pounds of work done in one minute. If an engine can deliver 33,000 foot-pounds of work in one minute it is rated as 1 horse-power; if 66,000, 2 horse-power; and so on.

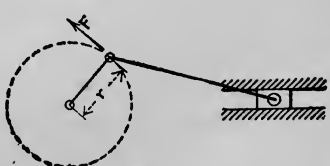


Fig. 19.

Generally engines deliver a rotative effort. Suppose, for example, that an engine transmits an average tangential force of F pounds at its crank pin. (See Fig. 19.) In one revolution, this force will act through a distance equal to the circle through which the crank pin has passed, or $2\pi r$, r being the radius of the crank in feet.

The foot-pounds of work per revolution are then $2\pi r \times F$, and if there are N revolutions per minute, the work done per minute will be $2\pi rF \times N$, and the horse-power will be

$$\text{H.P.} = \frac{2\pi rFN}{33{,}000}.$$

The quantity rF is called the " torque."

Dynamometers are used for measuring power. Generally the speed is measured independently of the dynamometer by a hand counter or otherwise, so the dynamometer is applied only to measure the torque. This is done by balancing the torque by a measured force acting at a known distance from the center of rotation.

There are two broad classes of dynamometers: *absorption* by which the power to be measured is converted into some form in which it cannot be used; and *transmission*, by which the power is passed on unchanged.

6. Constants of Friction Brakes

Principles. Friction brakes may be used as absorption dynamometers, and of these the **Prony brake** is the commonest and most accurate form. (See Fig. 20.) It consists of a band wrapped around a pulley on the shaft whose power is to be measured, so arranged that by tightening a hand wheel, H, the friction between the wheel and band can be controlled. The band is held from turning by means of an arm, A, attached to it, and supported by some force measuring device at its free end. Considering the friction between the band and wheel as a single force, f, then the horse-power developed is $2\pi r'fN \div 33{,}000$; r' being the radius of the wheel in feet. Now since the force W exerted by the scales, S, produces equilibrium, by the principle of moments

$$RW = r'f,$$

and therefore by substituting in the expression for horse-power just given

$$H.P. = \frac{2\pi RWN}{33,000} = BWN.$$

In practice, if the force, W, and the revolutions per minute, N, are measured, the horse-power may be calculated, the length of the arm being known. B is called the " brake constant."

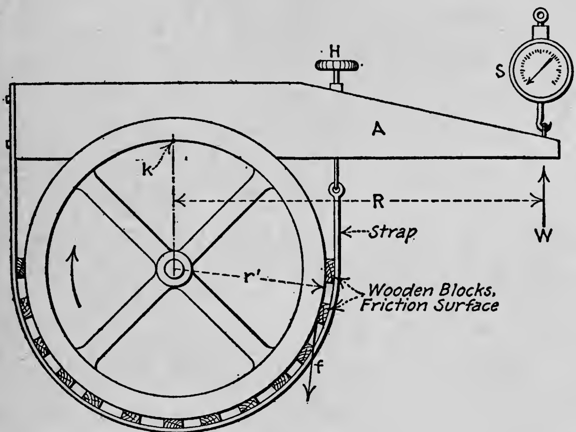


FIG. 20.—Prony Brake.

It should be noted that the arm, A, by its weight produces a force acting on the scales which should not be included in the force balancing the frictional effort. This should be allowed for by determining the " unbalanced weight " of the brake, or " brake zero," and subtracting it from the scale readings. Methods of determining the brake zero will be given later.

The rope brake is a modified form of prony brake. Fig. 21 shows such a one, the friction between the rope and the wheel being balanced by the force of the scales on the right less the weight on the left. The difference between these quantities is

the value of W in the formula, and the value of R is the radius of the wheel plus the radius of the rope. There is no unbalanced weight if the lengths of rope on each side of the pulley are equal. The friction, and therefore the horsepower, may be varied by increasing the weight on the left or by taking more turns of the rope about the wheel; either procedure greatly increases the friction.

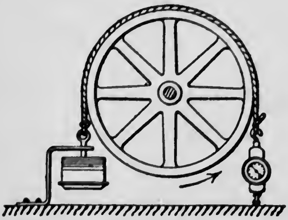
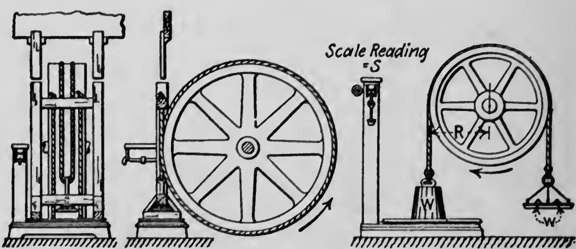


FIG. 21.—Rope Brake.

Fig. 22 gives two other arrangements of rope brake and their equations.

In operation, the heat generated by the friction is removed by circulating water within the brake wheel, the latter being provided with internal flanges for that purpose. It is generally



Torque = (S − Brake Zero)R Torque = $(W-S-w)R$

FIG. 22.—Rope Brakes.

sufficient to feed the water at a rate just equal to the loss by steaming.

Hydraulic friction brakes are the same as those just described in principle though quite different in detail. A series of discs mounted on the power shaft revolve in a casing filled with water under pressure, the casing being free to revolve about the shaft. The friction between the discs and the water causes a turning

effort upon the casing which is balanced by a force measuring device the same as with the common prony brake. Regulation is secured by varying the water pressure. The constants are the same as for the other types, but in most cases there is no unbalanced weight.

(a) Determination of Unbalanced Weight. If friction could be entirely eliminated between the band and flywheel, the unbalanced weight would be the reading of the scale, Fig. 20. But, no matter how loose the band is, there is always some friction between it and the wheel tending to hold up the brake arm when the wheel is stationary. If the brake is removed from the wheel and supported by a knife edge or circular pin at the point k, Fig. 20, the other end being supported by the scales as in operation, then the scales will indicate the unbalanced weight provided that the flexible band does not change in its weight distribution. This is one method of finding this quantity.

Another method consists in revolving the brake wheel first in one direction and then in the other, and noting the corresponding readings of the brake scales. With the wheel revolving clockwise, Fig. 20, the scales will indicate the unbalanced weight *plus* the force necessary to balance the friction at the band. Anti-clockwise, it indicates the unbalanced weight *minus* this force, since the friction is reversed. Calling the force balancing friction X,

$$\begin{array}{ll} & \text{Unbalanced Wt.} + X = \text{1st reading} \\ & \underline{\text{Unbalanced Wt.} - X = \text{2d reading}} \\ \text{Adding,} & \text{Unbalanced Wt.} \quad = (\text{1st} + \text{2d reading}) \div 2. \end{array}$$

When applying this method it is necessary that X remain constant. The band should be quite loose, the bearing blocks wet with oil, and the wheel turned at a uniform rate.

The third method is the same in principle as the second, but the brake is revolved instead of the wheel. The spring balance, Fig. 20, is first drawn upward giving the weight $+X$ reading; then the weight of the brake is allowed to draw it

down for the weight $-X$ reading. Note that if the spring balance is slanted out of its correct position relative to the brake, a component force will be indicated.

(b) Determination of the Horse-power per Pound of Thrust per Revolution. This is the "brake constant" or $2\pi R \div 33{,}000$. R should be measured with a tape or measuring rod. Note that R is the *perpendicular* distance from the center of the brake-wheel to the line of the balancing force W, Fig. 20.

Problem 6_1. What is the brake constant if $R = 5$ ft. $3\frac{1}{4}$ in.? If unbalanced weight of the brake is 2.5 lbs., what scale reading would be necessary to balance 40 H.P. at 160 R.p.m.? *Ans.*, .001; 252.5 lbs.

Problem 6_2. By the third method, the scales indicate 10 lbs. when the brake is pulled up. (Fig. 20.) Its weight is not enough for it to drop, consequently the balance is reversed and the brake *pulled* down, for the weight $-X$ reading. The scales then indicate 2 lbs. What is the unbalanced weight? *Ans.*, 4 lbs.

Problem 6_3. If the unbalanced weight is 5 lbs., where should weight be added to balance the brake, and how much?

Problem 6_4. Given an arrangement like Fig. 20 except that the wheel turns anti-clockwise and the spring balance is inverted. If the R.p.m. = 100, arm = 4 ft., unbalanced weight = 14 lbs., and the balance reads 20 lbs., what is the horse-power? *Ans.*, 2.59 H.P.

7. Calibration of a Fan Brake

Principles. Fan brakes are convenient for measuring the output of high speed motors that operate at variable speed, such as automobile engines. Fig. 23 shows such a one. The energy of the shaft is absorbed by imparting kinetic energy and heat to the air. With a fixed set of vanes the power thus absorbed varies as the cube of the speed. The capacity of the brake at a given speed of rotation can be changed in only two ways, namely, by changing the size of the vanes or their distance from the center of the shaft. The first method may be taken by swinging the vanes around on their arms so that the effective area resist-

ing motion is lessened. For the second method, the fastenings of the vanes to the arms may permit the desired radial variation.

This type of dynamometer is sometimes mounted on a frame independent of the engine shaft and driven by a belt. In this case, the belt losses and brake journal friction (more or less indeterminate quantities) are added to the resisting effort.

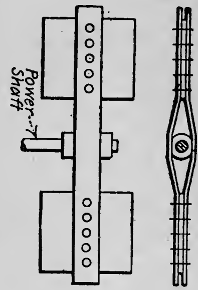


Fig. 23.—Fan Brake.

(a) Calibration against a Transmission Dynamometer. By means of a transmission dynamometer, simultaneous values of torque necessary to drive the fan and revolutions per minute are determined for a given adjustment of the vanes. From these values, the horse-power absorbed at each speed may be calculated, and a curve of horse-power vs. revolutions per minute plotted. This curve may be used for finding the horse-power, when the brake is in usual operation, from readings of the rotative speed.

(b) Calibration against a Calibrated Motor. If the fan brake is belt driven, it may be run by a variable speed electric motor. through the same belt and pulleys to be used when the fan measures power. At a definite speed of the fan, readings of the armature and field currents of the motor and of its speed are taken. The belt is then removed, a prony brake applied to the motor pulley, and the horse-power of the motor measured under the same conditions of current and speed. If all controllable conditions are the same, this horse-power will be the same as that absorbed by the fan at the applied speed. A series of similar trials at different speeds will give data for a calibration curve.

This calibration will be somewhat in error owing to the fact that the reaction on the motor bearing caused by the prony brake thrust is somewhat different from the reaction due to the belt pull; hence the friction of the motor under the two loads will be slightly different.

(c) Use of the Horse-power Constant. The power absorbed by the fan for a given setting of the vanes equals

$$K \times \text{density of the air} \times (\text{R.p.m.})^3$$

in which K is a quantity very nearly constant. Assuming that the density of the air also is nearly constant, the product of K and the density may be found experimentally at a convenient speed. Then the horse-power at any other speed may be found, approximately, by multiplying this product by the cube of the speed.

The experimental determination of K times air density is made by taking readings according to either method (*a*) or (*b*). Then the desired value equals the horse-power divided by the cube of the applied speed.

Variations in belt losses and in the value of K make this method inexact.

Problem 7_1. What percentage of error will be caused by the density of the air changing from that corresponding to a barometer of 29.5 in. of mercury, at which the fan is calibrated, to 30 in.? What percentage by a change of temperature from 60 to 80° F.? *Ans.*, 1.6%; 3.8%.

8. Calibration of a Transmission Dynamometer, Weight-arm Type

Principles. The weight-arm type of transmission dynamometer consists of some device by which the torque of a revolving shaft can be balanced by a standard weight or weights acting with a known leverage. The balancing torque caused by the weights is thus a measure of the horse-power when the revolutions per minute are known. There are many forms of this type of dynamometer differing mainly in mechanical details.

Belt dynamometers of different kinds are in the weight-arm class, of which Fig. 24 represents one. The torque of the trans-

mission shaft is equal to the effective belt pull, $T_1 - T_2$, multiplied by the radius at which it acts, r. Disregarding the friction at the bearings of the pulleys, p and p', the reactions at these bearings will be $2T_1$ and $2T_2$, respectively, as will be seen from a consideration of the belt forces. Taking moments about the fulcrum f, and disregarding the weight of the arm A, we have

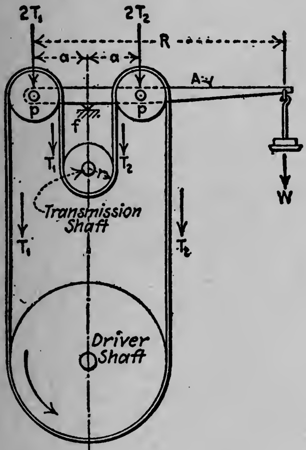


Fig. 24.—Belt Dynamometer.

$$WR + 2T_2 a = 2T_1 a$$

from which

$$T_1 - T_2 = WR \div 2a$$

and the torque

$$= r(T_1 - T_2) = rWR \div 2a.$$

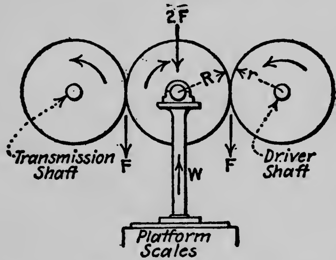


Fig. 25.—Pillow Block Dynamometer.

Thus by balancing the belt torque by a known weight W, the power may be measured if R/a is known and if a suitable allowance is made for friction.

A pillow block dynamometer is shown by Fig. 25, consisting of three gear wheels the middle one of which is mounted on a pillow block, so as to bear freely on a weighing device. The turning force of the driver, F, produces an equal resisting force at the gear on the transmission shaft, barring friction. The reaction W, which can be weighed by the scales, thus equals $2F$, and hence the torque of the transmission shaft is

$$Fr = \frac{W}{2} \times r.$$

The reaction W may also be measured by hanging the middle wheel from one arm of a lever above the other carrying a balancing weight.

Some forms of pulley block dynamometer omit the middle wheel and weigh the reaction of the bearing on the transmission shaft directly. When this is done, the transmission shaft must be arranged to rest freely on the weighing device.

The Webber differential dynamometer is a variation of Fig. 25, but similar in principle. It is shown diagrammatically by Fig. 26, which represents three bevel gears, the middle one being

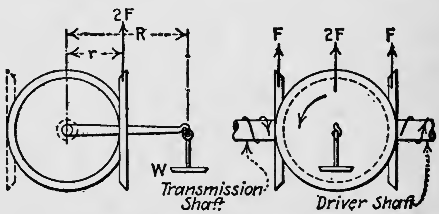


FIG. 26.—Webber Dynamometer.

supported by a lever whose fulcrum lies on the common axis of the driver and transmission shafts, Similar to the principle shown by Fig. 25 the net force on the middle gear is $2F$. This is balanced by the dead weight W acting with a lever arm of R feet. Taking moments of the forces acting on the lever around its fulcrum, we have

$$2Fr = WR$$

from which, torque transmitted =

$$Fr = \frac{W}{2}R.$$

As actually constructed there is another gear on the lever-arm, as indicated by the dotted line. This does not change the relation, merely substituting for the moment $2Fr$ two moments,

$Fr + Fr$. Also there is a jockey weight on the arm in addition to the pan weights similar to the arrangement of a platform scales.

The Emerson power scale is a dynamometer by which the torque is passed to a transmission pulley on the driver shaft through a lever system so arranged that a thrust in an axial direction is given to a non-revolving sleeve on the shaft. This thrust is balanced by dead weights which consequently measure the torque.

Friction of the moving parts of the dynamometer itself has not been accounted for in the equations. Generally the dynamometer indicates the torque necessary to overcome its own fricton and windage in addition to the external torque on the transmission pulley which alone it is the purpose to measure.

(a) The Constants of a Dynamometer of the Weight-arm Type are first, the true values of the dynamometer weights; second, the unbalanced weight of the arm or lever system; and third, the leverage ratio produced by the arm or lever system.

The true values of the weights should be determined with a scales sufficiently precise to keep within a reasonable percentage of error.

The unbalanced weight of the arm may be found as for a prony brake, by revolving the driver shaft in first one direction and then the other by hand, and noting the resulting force at the end of the arm where the weights are to be applied. A spring balance may be used for this purpose. Half the sum of the two readings equals the unbalanced weight.

The ratio of levers may be obtained generally by direct measurement. In the case of the Webber dynamometer, for instance, only the length of R, Fig. 26, is necessary. For the arrangement of Fig. 24, the ratio is $R\ a$, as shown by the equation.

(b) Calibration by Calculation. The torque equivalent to each dynamometer weight acting at the previously determined leverage is found by multiplying its value in pounds by the lever-

age ratio as shown in the equation for the dynamometer in question. A series of such determinations furnishes data by which the horse-power corresponding to any weight may be found when the dynamometer is in usual operation.

The torque equivalent to the unbalanced weight of the arm should be added to that indicated by the weights applied if the turning effort of the driver on the arm is upward. If it is downward, the power being then gaged by scales instead of weights simply, the torque equivalent to the unbalanced weight of the arm should be subtracted.

(c) **Allowance for Friction, Windage, and Centrifugal Force.** To allow for friction, a crude but convenient approximation is based upon the assumption that the frictional resistance is the same under all conditions of external torque. If, then, the dynamometer is run with the transmission shaft entirely free, the weight to balance the arm gives the correction to be subtracted from the readings in usual operation. Windage is included in this, but if the dynamometer is to be used at various speeds, similar corrections should be determined at these speeds to allow for the varying value of the windage.

With some dynamometers having revolving levers, bell-cranks, and the like, such as the Emerson power scale, centrifugal force acting on these parts may cause a distortion of the indications. This is allowed for under the windage corrections.

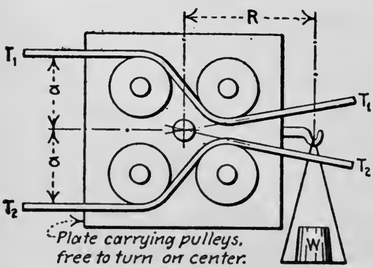


Fig. 27.—Belt Dynamometer.

(d) **Comparison with Prony Brake Measurements** is the most valid means of calibrating any transmission dynamometer. To do this, a prony brake is fitted to the transmission pulley and is adjusted so as to balance each dynamometer weight in turn, power being delivered as in usual

operation. The torque shown by the prony brake is the true torque measured by the corresponding weight. When a jocky weight is used as for the Webber dynamometer, a calibration curve of torque against jockey weight positions is convenient.

If the dynamometer is to be used at various speeds, it should be calibrated at a number of them, the results to be in terms of either horse-power or torque, as shown by the brake, corresponding to each dynamometer weight. In this way friction, windage, and centrifugal force may be taken into account.

Problem 8_1. Deduce the torque equation for Fig. 27.

9. Calibration of a Transmission Dynamometer, Spring Type

Principles. Spring dynamometers differ from the weight-arm type in that the torque to be measured is balanced by a a spring or springs through which the torque is passed. The spring is consequently deformed by either a tensile, compressive, or twisting stress. If the constant of the spring is known (that is, the number of pound-feet of torque necessary to cause a unit deformation) then by noting the deformation, the horse-power may be determined.

Fig. 28 shows, in part, the principle of the **Van Winkle dynamometer.** Power is taken off at the loose pulley P which is driven through springs by the disc attached to the driver shaft. The resulting deformation of the springs permits a change of position of the pulley relative to the disc, and this operates a bell-crank lever (not shown) which in turn actuates a pointer on a stantionary dial. The dial is arranged to indicate horse-power direct.

Another spring **dynamometer, made by the Central Laboratory Supply Company,** is shown by Fig. 29. Two shafts are connected by a spring through which the power to be measured is passed. These shafts are provided with discs arranged as commutators,

being insulated from the shafts except at the shaded portions shown under the brushes. In this position, an electric circuit is

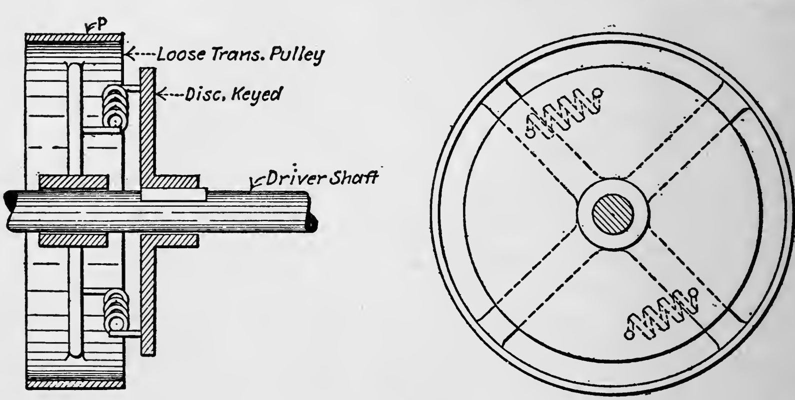


Fig. 28.—Van Winkle Dynamometer.

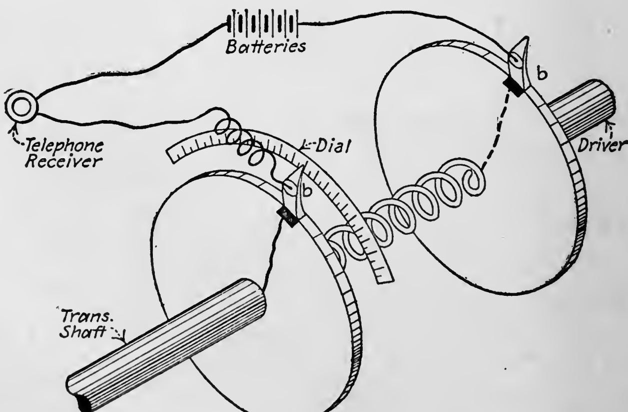


Fig. 29.—Central Laboratory Dynamometer.

completed causing a click in a telephone receiver. The right-hand brush is stationary, but the one on the left is so arranged that it may be swung around the shaft. In operation, the latter is

manipulated until a click is heard in the receiver, indicating that both contact pieces are passing under the brushes at the same time. Then the angular motion of the left-hand brush shown on the dial and measured from its clicking position when there is no torque delivered, is equal to the twist of the spring, and hence is a measure of the torque.

In usual operation small variations of the torque, due to belt flapping, etc., make the clicking position somewhat variable. It is therefore convenient to read the maximum and minimum angles at which no click is heard and to take the average of these as the twist of the spring.

The Flather dynamometer (Fig. 30) differs from the foregoing in that the torque is communicated to the transmission pulley through small pistons, *p*, working in cylinders filled with oil.

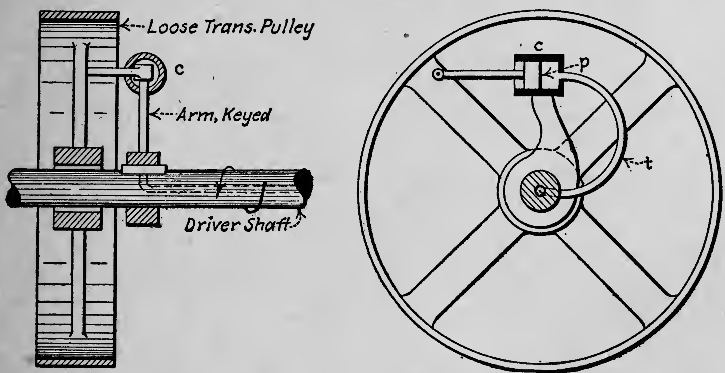


Fig. 30.—Flather Dynamometer.

The fluid pressure thus produced is transmitted through tubes, *t*, and a longitudinal hole through the center of the driver shaft. Connecting with this hole at the end of the shaft is an indicator by which a graphic record of the pressure is made. Since this pressure is proportional to the torque (being produced by the driving force at the cylinders, *c*, acting at a constant distance from the shaft center) it is a measure of the horse-power.

Friction affects the indications of a spring dynamometer in two ways. First, since the friction of the loose transmission pulley or of the transmission shaft on its bearings always acts *against* the motion, it makes the indicated torque *greater* than the true external torque. Second, friction of the indicating mechanism makes the indications for increasing torques low, and for decreasing high, similar to the action of friction in a pressure gage. Generally the first effect is greater than the second.

(a) **Static Calibration.** With the driver shaft clamped, the transmitting shaft may be twisted with a series of known weights acting at a measured distance from the center of the shaft. Thus a series of values of torque may be obtained with the corresponding instrument indications, the latter being either in dial graduations, degrees of twist, or height of an autographic diagram, as may be appropriate to the instrument tested. To apply the static torque, the transmission pulley may be used as the arm, a rope being tied to a spoke and passed over the pulley face from which to hang the weights. If the necessary number or size of standard weights are not available, a lever may be clamped to the transmission shaft, and a single weight applied at various distances from the center. In this case, the moment of the lever should be accounted for.

Increasing and decreasing values of the torque should be applied and corresponding readings taken to eliminate the effect of friction. For the increasing readings, the weights are caused to bring up the torque gradually to the desired amount. For decreasing, extra torque is brought upon the shaft by bearing on the weights by hand; then gradually removing the hand pressure so that the torque will decrease to the desired value. The average of each pair of readings is then plotted against torque for a calibration curve. By this procedure, for increasing values, the motion of the transmission pulley or shaft is opposite to that for decreasing values. Hence both effects of friction, previously noted, are eliminated.

For dynamometers with which the angle of torsion is read, the curve of torque vs. degrees may be used to get the spring constant. Then if S is this constant in pound-feet per degree of twist and A the torsion angle noted in usual operation,

$$\text{horse-power} = .00019 \times S \times A \times \text{R.p.m.}$$

which is the horse-power delivered by the dynamometer pulley.

(b) Allowance for Friction, Windage, and Centrifugal Force. The readings of the dynamometer should be reduced by an amount corresponding to the torque necessary to overcome these forces. The values of the corrections may be determined as for weight-arm dynamometers. For recording instruments, a line should first be made on the chart with the dynamometer running free. The diagram under load should then be measured from this friction line as a datum.

For dynamometers using a measurement of the torsion angle, the correction is made in degrees, being subtracted from the reading observed in operation.

(c) Comparison with a Prony Brake may be made in exactly the same way as for weight-arm dynamometers. (Test 8 (*d*).)

Problem 9_1. The reading of a spring dynamometer running at 450 R.p.m. is 26 degrees. If the spring constant is 3 lb.-ins. per degree, and if the correction for windage, friction, etc., is 5 degrees, how many foot-pounds of work will be done in ninety seconds? What will be the horse-power transmitted? *Ans.* 22,200 ft.-lbs.; 0.45 H.P.

Problem 9_2. Same data as Problem 9_1, except that the friction of the transmission shaft is separately determined, the correction being 4 degrees, and the correction for friction of the indicating device is 1 degree. What is the transmitted horse-power when the torque is increasing? When it is decreasing?

Problem 9_3. The axis of the piston p (Fig. 30) is 15 inches from the center of the driver shaft, and the diameter of the piston is 2 ins. Disregarding friction, what fluid pressure is produced when the horse-power is five, at 150 R.p.m.? *Ans.* 44.0 lbs. per sq. in.

THE ENGINE INDICATOR—REDUCING MOTIONS

The engine indicator is an instrument which makes a graphic diagram giving the relation between the pressure and volume of the fluid in an engine cylinder under working conditions. Since the area of such a diagram is proportional to work, the indicator is a dynamometer of a special type. It is shown in principle by Fig. 31. *C* is a small cylinder with a close fitting piston *P* which is subject to the same fluid pressure as in the engine cylinder, being connected to it by a short pipe. This pressure compresses the spring *S* until the force of the spring balances the pressure. Thus the motion of the piston is proportional to the pressure. The piston motion is communicated and magnified by means of a linkage *L* bearing a pencil point at *A*. *D* is a metal drum free to oscillate on a spindle and carrying the paper on which the record is to be made. The drum is actuated by the engine cross-head through a cord *K*, a spring within the drum serving to bring it back upon the return stroke. Thus the motion of the drum is proportional to the engine piston and therefore to the volume in the engine cylinder behind the piston. The diagram made by the pencil point on the record paper is one of pressure shown vertically and volume (or piston stroke) horizontally.

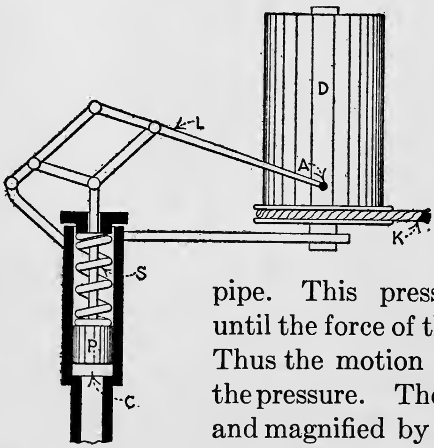


Fig. 31.—Engine Indicator.

The principal use to which the indicator is put is the finding of the *mean effective pressure* in an engine cylinder throughout its working stroke, from which quantity the *cylinder or*

indicated horse-power may be calculated. (See Test 43.) Fig. 32 is a typical indicator diagram from a steam engine. The average pressure on the forward stroke is the mean height, to scale, of the curve *abc*. On the return stroke, the average pressure (*back* pressure) is the mean height of *cde*. The effective pressure is the difference between these two, or the mean height of the indicator diagram. This may be found by dividing its area in square inches by its length in inches, and multiplying the quotient by the scale of pressure. The scale of pressure is

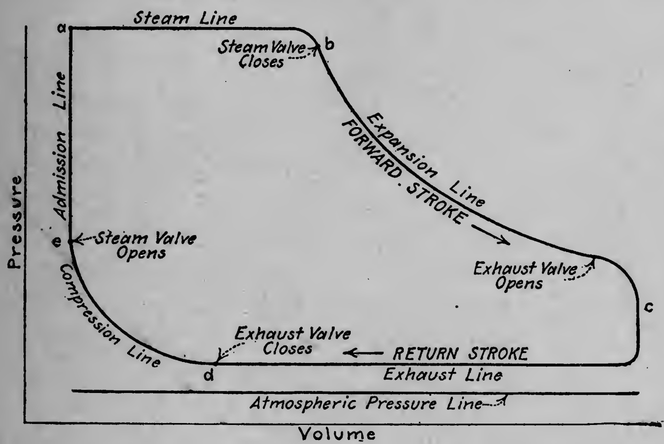


Fig. 32.—Indicator Diagram from Steam Engine.

the number of pounds per square inch on the indicator piston necessary to produce one inch of rise of the pencil, and is referred to as the *spring scale*.

The indicator is equipped with a number of springs of different stiffnesses so that one appropriate to given conditions may be selected.

The drum spring is adjustable so that a greater tension may be operative at higher rotative speeds to provide proper acceleration of the drum upon the return stroke.

The cord actuating the drum does not receive its motion directly from the engine cross-head, but from a *reducing motion*, the function of which is to reproduce the engine stroke on a small, but always proportionate scale. A pantagraph is often used for this purpose. Under Tests 12 and 13 are described various types.

10. Calibration of the Indicator Spring and Pencil Motion

Principles. There are four causes of error in the ordinates of an indicator diagram. First, when applied to high speed engines, the inertia of the indicator piston and attached linkage causes a variation from correct positions. Second, at high speed, the pressure in the indicator cylinder lags behind that operating on the engine piston, because of the inability of the steam immediately to traverse the passages to the indicator. Third, the mechanism actuating the pencil may incorrectly magnify the motion of the indicator piston. Fourth, the spring scale may not be exactly known. Friction of the indicator piston and linkage causes a variation of the spring scale, since because of it the pencil is too low when rising and too high when falling. (See Test 2, principles.)

The first of these errors may be avoided, or reduced, by the use of stiffer springs than are appropriate to low rotative speeds. With a stiffer spring, the total rise of the pencil is less, and therefore the velocity and inertia of the pencil motion parts are decreased.

The second cause of error, lag in the fluid pressure, may be reduced by the use of short and direct pipe connections between the indicator and engine cylinders.

Errors due to the third and fourth causes may be corrected as will be described.

(a) The Pencil Motion may be Tested as Follows: With the indicator spring removed, a horizontal line is drawn on a piece of record paper placed on the drum, by revolving it by hand, the pencil bearing against the paper at a low position of the linkage. Then, with the drum held stationary, a vertical line is made by moving the pencil and linkage up by hand. This is repeated with the drum in a second position. The two vertical lines should be parallel and straight, and perpendicular to the horizontal line, if the pencil motion is true.

(b) Determination of Ascending, Descending, and Combined Spring Scales by Graphic Method. It is necessary to get a series of values of true pressures and corresponding heights of indicator pencil. The apparatus for varying and measuring the pressure should preferably be one using the same working fluid to which the indicator is subjected in practice, so as to duplicate the conditions of temperature and friction. Dead weights applied directly to the indicator piston are sometimes used, but these do not correctly reproduce the working conditions. Fig. 33 represents a calibration apparatus using steam. The pressure is varied by manipulating valves *A* and *B*, an opening of *A* and closing of *B* having the effect of increasing the pressure in the large steam chamber, and vice versa. The measuring device is a set of known weights acting on a plunger of known area, from which the pressure balancing the weights may be figured. An accurate and precise Bourdon gage would serve the purpose as well. The gage shown in Fig. 33 is used to indicate the pressure in the chamber when the weights are not balanced, for convenience in manipulating the valves *A* and *B*.

Using the calibration apparatus, a diagram similar to Fig. 34 is made on an indicator card. For the ascending pressures, the weight table must be balanced and rising very slowly when the line is drawn. Similarly, for descending pressures, the table should be gently falling. The table is revolved by hand to reduce friction at the plunger.

The heights of the lines from the atmosphere line are then measured to $\frac{1}{100}$ in., and recorded on the diagram with corresponding pressures as in Fig. 34. These data are then plotted and results obtained as for Test 2 (*a*).

(c) **Spring Scales by Method of Least Squares.** The same experimental data are used, but the results are calculated as for Test 2 (*b*), the value of F in the equations then being the observed

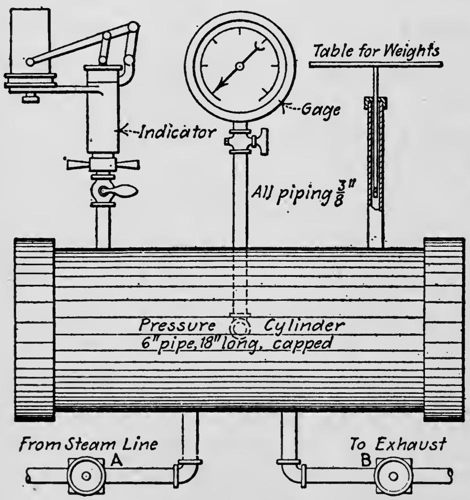


FIG. 33.—Indicator Spring Testing Apparatus.

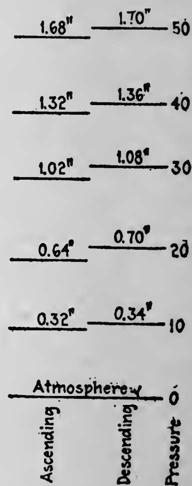


FIG. 34.—Calibration Records.

pressure in pounds per square inch, and E, the height of the indicator pencil in inches.

(d) **Correction of Indicator Diagrams.** Various methods, more or less accurate and laborious, have been proposed for applying calibration results to indicator diagrams. Generally, it is sufficient to use the combined spring scale with which to multiply the mean height to get the mean effective pressure, the error involved being within the limit of accuracy of power tests. But the combined spring scale represents the true scale of the spring

more nearly than it does actual conditions, since the method of figuring it eliminates friction. Strictly speaking, the ascending and descending scales should be used separately on the diagram, the former applied to the mean height of *cde*, Fig. 32, since the pencil is rising on that line; and the latter to the mean height of *abc* since there the pressure falls. But when the back pressure line is horizontal in large part, as it almost always is, the descending scale applied to the whole diagram will yield a fair result.

The indicator is sometimes applied to other than power measurements, for which a high degree of accuracy is desirable.

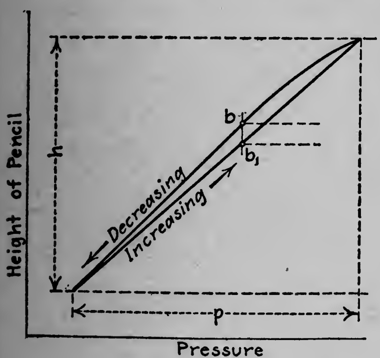


FIG. 35.

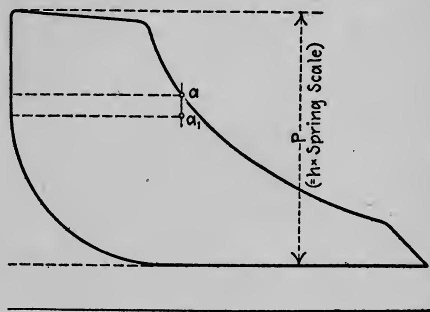


FIG. 36.

Correction of Indicator Diagrams.

It is then necessary to reconstruct the diagram to get correct results. A method of doing this is as follows. Calibration data are obtained as previously described, and plotted as shown by Fig. 35. The range of pressure P for the calibration must be the same as that in the indicator diagrams to be corrected, and all operating conditions of the indicator during its test the same as when the diagrams are taken. It will be noticed that the descending curve of Fig. 35 is not straight, and is joined to the ascending curve. Careful experimentation will reveal these characteristics. Actually, there can be no unfilled gap between

the two curves, as there would be under the assumption that both are straight lines, an assumption, as was pointed out under Test 2, made only for convenience in approximately figuring the scales.

Having plotted the curves according to Fig. 35, the *ascending* scale should then be obtained by either of the two methods (*a*) or (*b*) previously given. The calibration curve is next laid alongside of the indicator diagram to be corrected as shown by Figs. 35 and 36. Now, the back pressure line is shown correctly to the ascending scale determined from the calibration. To represent a point a of the indicator diagram on this scale, the construction abb_1a_1, is used, the point a_1, being the required corrected position of a. Enough points are corrected in this way to reconstruct the pressure line of the indicator diagram. The ascending scale will then apply correctly to the whole diagram.

(e) **Sampling.** When there are a large number of diagrams for a single engine test, a few representative ones only are selected for reconstruction. Suppose, for instance, that there are 24 diagrams and that the average of all their mean heights is H inches. Three of these should be selected as near as possible to this average mean height, and fairly representative in general proportions. These three diagrams are reconstructed and then the desired results obtained from them.

To get the average of the mean effective pressure from all the diagrams without reconstructing all of them, the value H may be reduced in the proportion that the mean height of the three sample diagrams is reduced after reconstruction. The corrected value of H when multiplied by the ascending spring scale gives the corrected mean effective pressure.

Problem 10_1. The pencil of an indicator throws 2 ins. If the boiler pressure is 145 lbs., what should the spring scale be to give as high a diagram as possible? *Ans.*, 80.

Problem 10_2. The ratio of the pencil motion to the piston motion of an indicator is 6 : 1. The area of the piston is $\frac{1}{2}$ sq. in. With a 30-lb. spring, the difference between ascending and descending positions of the pencil at

a given pressure is 0.04 in. If the friction is all at the piston, how much is it in pounds? *Ans.*, 0.3 lb.

Problem 10_3. With the same indicator and spring at last problem, if the friction between the pencil point and the paper is 2 oz., how much difference in the height of the pencil will this make? *Ans.* 0.05 inch.

Problem 10_4. With the apparatus of Fig. 33, what is the effect of friction at the plunger upon the apparent true pressure? What effect has this upon the calibration records?

11. Testing the Motion of the Indicator Drum*

Principles. When an indicator drum is driven by a cord attached to a reducing motion mechanically correct, the motion imparted to it is approximately simple harmonic. Upon the forward stroke of the engine, the cord is pulling the drum, and upon the return stroke, the drum spring is stressing the cord to keep it taut. Thus the cord is always under stress. If the stress varies, the cord will stretch according to such variation, and consequently the drum will not assume the correct positions it would have if driven by an inelastic connector.

Stress in an indicator cord is the resultant of three distinct forces: namely, the drum spring tension, the force required to overcome the inertia of the drum, and the force to overcome friction at the drum spindle and at any guide pulleys used to guide the cord. The force of the spring increases with the forward stroke of the drum (the spring being wound up) and decreases upon the return. The variation is usually uniform with the motion.

In general, at the beginning of the forward stroke, inertia increases the cord stress since, as the speed is increasing from zero to a maximum at mid-stroke, inertia effects a tendency of the drum to lag. Beyond mid-stroke, however, the speed is decreasing, and as the drum tends to exceed the velocity induced

*For a more complete discussion of this subject see author's article in *Power*, Aug. 20, 1912.

by the cord, a slackening results. Upon the return stroke the force of acceleration varies in the same way as on the advance.

The force of friction is always opposite to the drum motion and therefore changes its direction at each stroke. During the forward stroke it tends to increase the cord stress, and upon the return, to decrease it.

These three forces are represented graphically by Fig. 37. The resultant stress in the cord equals their algebraic sum at any drum position. At low speeds, the force of acceleration

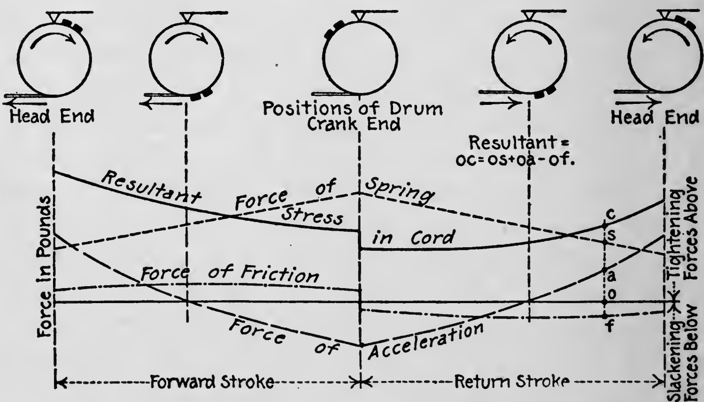


FIG. 37.—Variation of Forces on Indicator Card.

is practically negligible; hence the resultant cord stress increases on the forward stroke, and decreases on the return with lowered values due to the reversal of friction. At high speeds, the force of acceleration reverses this variation as shown by Fig. 37. At some intermediate speed the force of acceleration and the spring tension nearly balance, so that the cord stress is approximately constant during each stroke, but a trifle less on the return because of friction. This is the speed at which the indicator drum is best adapted to work, since the more nearly constant the cord stress is, the less is its stretch and the consequent error.

Let us now compare the effects of a higher speed than this with the ideal condition of constant cord stress. The effect of speed is to increase the stress at the head end and decrease it at the crank end (see Fig. 37). At the head end, therefore, the cord is longer and the drum travels further in this direction. Similarly, at the crank end, the cord is shorter and the drum is pulled further toward that end. The net effect is to pull out the indicator diagram. Now, if the diagram were lengthened *uniformly* its proportions would remain correct, and there would be no error. Accordingly, if the cord were subjected to a uniformly decreasing and increasing stress, and if its stretch varied directly with the stress, a correct diagram would result. It follows that, lacking such uniformity, the error of any point of the diagram should not be judged by the absolute stress or stretch at that point, but by the difference between this stress or stretch and that necessary to produce a uniform lengthening. It is seen from Fig. 37 that the resultant cord stress is greater than that necessary for uniformity on the forward stroke and less on the return. Hence, during the forward stroke, the cord is too long and a point on the indicator diagram is to the left and behind its correct position. During the return stroke the cord is too short; a point on the indicator diagram is to the right and again behind its correct position, since the motion is reversed. The net effect of the stretch of the cord, then, is to make the mean effective pressure appear smaller, and the cut-off, compression and release earlier than their true values.

(a) **Determination of Spring Tension.** Since this varies through the drum stroke, its value at mid-stroke may be measured, or the average of the values at the ends of the stroke may be used. A spring balance of between 5 and 10 lbs. capacity should be fastened to the cord leading from the drum, and through it the drum pulled past the position at which it is desired to measure the spring tension. The reading of the balance is noted at the instant of passing this position. This equals the spring tension

plus friction. The spring is then allowed to reverse the motion, and another reading is taken, equal to the tension minus friction. The sum of these readings divided by two is the spring tension.

(b) **Determination of Drum Friction.** The same method is used as for measuring the spring tension. The friction equals the *difference* of the readings divided by two.

With an indicator properly lubricated and adjusted, and of good make, the friction should not exceed 5 or 10 per cent. When the cord runs over guide pulleys, however, the friction is greatly increased. When the friction is high, the cause should be looked for and remedied.

(c) **Testing Indicator Cord.** Tie one end of about four feet of the cord to be tested to a fixed point on a bench or table, and the other end to a spring balance. Mark this end a few inches from the balance with a fine ink line, and under this line place a piece of paper or a foot-rule. Stretch the cord by pulling the balance horizontally until about 5 lbs. are indicated. Now reduce the force to about 1 lb. and repeat this procedure a few times. The elongation may then be noted for the applied range of stress. As the cord in the operation of the indicator is generally not stressed more than 5 lbs. or less than 1 lb., this range is appropriate. A good cord should not stretch more than 0.01 in. per foot per pound, but grades will be found with four times this stretch and more.

The elongation and contraction of the cord are very much greater at stresses less than 1 lb. On this account there may be marked overtravel at the crank end of the drum motion without a visible slackening or vibrating of the cord when the indicator is in usual operation.

(d) **Adjustment of Drum Spring Tension.** The indicator is put in working adjustment on the engine to be indicated, the piston spring omitted. Then by putting the engine on first one dead center and then the other, two vertical lines may be made on an indicator card to mark the extremes of travel of the

drum. When the engine is run, a horizontal line will overtravel the vertical ones, as was previously demonstrated, if the speed is enough to cause any inertia effect. The overtravel at the crank end should not be greater than 0.01 in. for each foot of cord, the cord being of good quality. If the overtravel exceeds this, the spring should be tightened, until it is reduced to the named amount.

The spring should not be any tighter than necessary, as this would increase the reaction on the drum spindle and therefore the friction. For different speeds, different spring tensions should be used.

(e) Testing Drum Motion with the Drum Motion Tester. The apparatus shown by Fig. 38 was devised by the author for

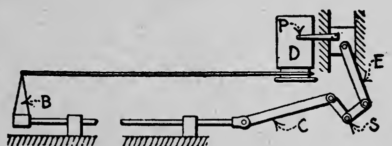


Fig. 38.—Smallwood's Drum Motion Tester.

this purpose. A shaft *S*, the rotary speed of which may be controlled by a variable speed motor, actuates two similarly proportioned crank trains *E* and *C*, set exactly 90 degrees apart. The motions of the two cross-heads are thus exactly proportional at all parts of their strokes. The indicator drum *D* is oscillated by fastening its cord to a bracket *B* carried on an extension of the horizontally moving cross-head. The cross-head with vertical travel carries a pencil point *P* which traces a diagram on a card on the drum. A diagram thus obtained is one of cross-head motion shown vertically and drum motion horizontally. If a rigid connector between the drum and the bracket were used instead of the indicator cord, the motion of the drum would be exactly proportional to that of the pencil, and the diagram

would be an inclined straight line. The effect of an elastic connector, as cord, variously stressed, is to give a curved line. If a straight line is drawn between the highest and lowest points of this curve, then the horizontal departure of any point on the curve from the straight line shows the error in the drum motion at that point.

When testing a drum motion for errors in an indicator diagram previously obtained, great care should be taken to reproduce all the operating conditions, relative to speed, spring tension, length of diagram, length of cord, general adjustment, and

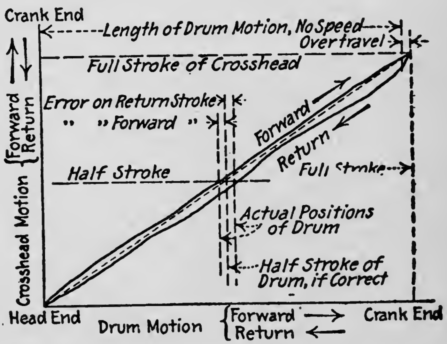


Fig. 39.—Error Diagram from Drum Motion Tester.

friction. Special care should be taken to reproduce the arrangement of guide pulleys. The effect of guide pulleys, especially if near the indicator, is similar to that of drum spindle friction and imposes an additional tightening force on the forward stroke and slackening one on the return. Fig. 39 shows a typical error diagram taken with the drum motion tester and is self explanatory.

In some cases the drum advances its correct position instead of lagging behind it on the return stroke. This is caused by the departure of the acceleration force from the variation assumed in Fig. 37, because of abnormal stretch of the cord and the consequent change in the drum velocities.

(f) The Correction of Indicator Diagrams from Drum Motion Tester Records may be accomplished as shown by Fig. 40, which needs no comment. Sampling of the diagrams may be done as indicated under Test 10 (*e*).

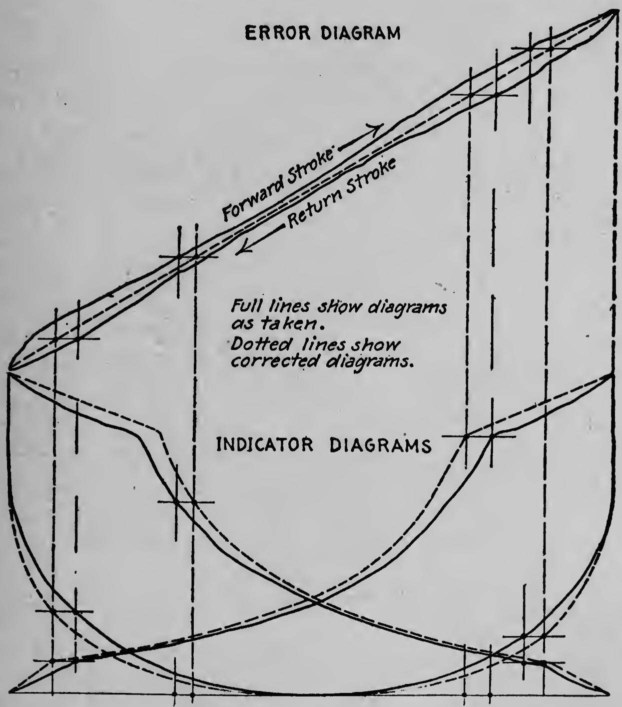


FIG. 40.—Correction of Indicator Diagrams.

Problem 11_1. The force of acceleration of the indicator drum varies directly with its stroke and the square of the engine R.p.m. If a drum motion with a 4-in. stroke is designed correctly to operate at 200 R.p.m., how long should the stroke be to operate correctly at 250 R.p.m.? At 300 R.p.m.? *Ans.*, 2.56 in.; 1.78 in.

Problem 11_2. What should be the constant of a drum spring (pounds per inch of extension measured on the card) to balance throughout a 4-in. stroke a force of acceleration which has a maximum value of +1 lb. at the

head end, and −1 lb. at the crank end? If the engine R.p.m. is doubled, what should be the constant of the spring? *Ans.*, 0.5 and 2.0 lbs. per in.

12. The Testing of Link Type Reducing Motions

Principles. The pantagraph in various forms has been much used to reduce engine cross-head motion for the purpose of driving indicators. Fig. 41 shows a number of them, diagrammatically. In these and in the following two figures, the letter F denotes the fixed center of the linkage; R, the point at which the indicator cord is attached; and C, the point of attachment of the linkage to the engine cross-head.

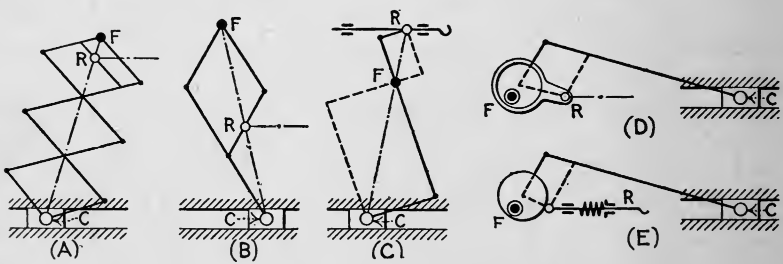


Fig. 41.—Pantagraph Reducing Motions.

There are two conditions necessary to the proportionality of motion of the points R and C. First, they must lie on a straight line passing through F, and second, the links that are parallel in one position must remain so in all positions.

Fig. 41 A and B are familiar forms, the reduction of motion being in the proportion of FR to FC. The identity of Fig. 41 C may be established by the dotted lines, the linkage thereby represented being replaced by the sliding bar on which the point R is centered. With this arrangement, the ratio of the long to the short link below F must be equal to the corresponding ratio above, in order to fulfill the condition of parallelism. This reducing motion is appropriate to high speed engines since by it only

a short length of cord need be used. Fig. 41 *D* is a convenient form of pantagraph made by attaching to the engine crank shaft a small eccentric and rod, or crank and rod, the throw of which is equal to the desired drum stroke. It will be seen that for a correct reduction, the eccentricity must be in line with the engine crank; and the ratio of the lengths of eccentric rod to eccentricity must equal the ratio of the engine connecting rod to crank length. Fig. 41 *E* is a modification of this, the motion being the same as that of a Scottish yoke, that is, a crank train with infinitely long connecting rod. The motion is therefore inaccurate.

In each case, the proportional motion of *R* is *parallel* to that of *C*. Therefore, the indicator cord should be led from the reducing motion parallel to the cross-head guides; if on a slant, the drum motion will not be proportional to the cross-head motion.

Reducing motions of the pendulum type are represented by Fig. 42. By *A* is shown the slotted pendulum. The shorter

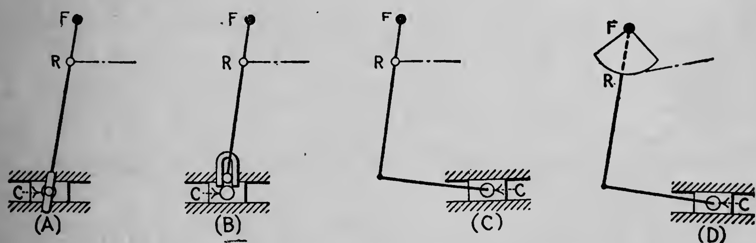


Fig. 42.—Pendulum Reducing Motions.

is the indicator cord, the greater will be its angularity due to the circular motion of *R*. The motion of *R* is not truly proportional to *C* since the ratio of *FR* to *FC* varies. Fig. 42 *B* shows the slotted cross-head, a correct motion except for the angularity of the cord. Fig. 42 *D* is the same as *C* except that a " brumbo " pulley is attached, by which the cord may be led off at an angle with less error than if the pulley were not used. With a horizontal

cord, the pulley causes more error than would be obtained without its use.

The errors of these reducing motions are kinematic and mechanical. Mechanical errors are due to lost motion in the joints, or flexing of the links. Kinematical truth or errors depend on the design.

(a) **Calibration by Line Diagram.** The crosshead motion is laid out to scale as shown by XY, Fig. 43, and divided into a number of equal parts. The centers of the reducing motion are then located to represent it in an extreme position, and a point D to show the corresponding position of the indicator drum. A second position of the reducing motion is then drawn and a second drum position marked, and so on. It is then an easy matter to measure the error of any of the drum positions since the distances between them should be equal for exact proportionality of motion. If desired, the data may be plotted the same as Fig. 39, from which indicator diagrams may be corrected according to Test 11 (*f*).

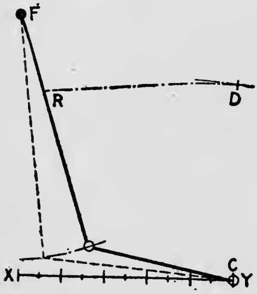


Fig. 43.

(b) **Calibration by Direct Measurement.** The indicator and reducing motion are set up as in actual use. The dead center positions of the engine cross-head are marked on the cross-head guides against a datum line on the cross-head. With the cross-head placed at any part of its stroke, its position may then be readily measured from either dead center and expressed in per cent of the stroke. For a proportional motion, the indicator drum should have moved from its corresponding dead center position (located by marking with the indicator pencil a vertical line on a card on the drum) the same percentage of its stroke. A number of such measurements at different parts of the strokes furnish data which may be applied the same as under (*a*).

Care should be taken that motion is not lost through stretch of the indicator cord. For reliable results, wire should be used between the reducing motion and the indicator drum. If the wire is not sufficiently flexible to pass around the drum, a few inches of cord tied to the wire may be used for this purpose.

Problem 12_1. The stroke of the drum given by the reducing motion of Fig. 41 *E* is 4 ins. The ratio of the engine connecting rod to crank length is 5 : 1. Figure from the kinematic formulas the maximum error in the drum motion.

Problem 12_2. Is the best arrangement of Fig. 42 *C* with the lower link always above the horizontal, always below it, or partly above and partly below? Why? What effect has the length of this link upon the accuracy of the motion?

Problem 12_3. Draw two indicator diagrams, superimposed, to show the effect of twisting the eccentric of Fig. 42 *D* ahead of the crank by 10 degrees.

13. The Testing of Reducing Wheels*

Principles. Link reducing motions have been largely supplanted by reducing wheels on account of the latter's ready adaptability. This type of motion consists, in general, of two drums of different diameters mounted on separate shafts that are connected together with gears. Fig. 44 shows them on the same shaft for simplicity. The indicator cord is led from the engine cross-head to the larger drum to which it is fastened. Another cord connects the smaller drum to the indicator drum. It will be seen that the velocity of the one cord is to the other as the ratio of the drum diameters of the reducing motion. The reducing wheel is supplied with a spring which acts the same as the indicator drum spring. One of the reducing motion drums is made interchangeable with others of various diameters so that different reductions may be made.

* For a more complete discussion of this subject, and experimental results, see author's article in *Power*, Sept. 24, 1912.

In the operation of a reducing wheel the same forces are at work as in the case of the indicator drum (Test 10, principles), namely, spring tension, friction, and the force due to inertia. Since the indicator drum and wheel masses are connected by a short cord having inconsiderable stretch, they may be regarded as one mass producing a single inertia effect. Likewise, the

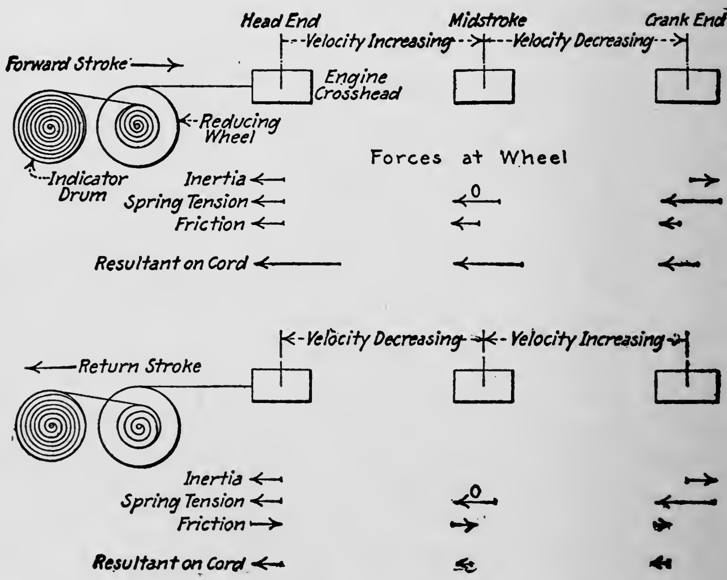


FIG. 44.—Forces on Cord Driving Reducing Wheel.

two springs may be considered as exerting a single force. In Fig. 44, the component forces are shown as they are felt at the cord at three parts of each forward and return stroke. The arrows represent roughly by their length the comparative magnitude of the forces.

At the beginning of the forward stroke, the cord exerts a maximum force in overcoming friction, inertia and spring ten-

sion, but as most of the cord is wound on the wheel, its stretch and the consequent lag of the drum are inappreciable.

As the cross-head advances, the length of cord free to stretch becomes greater, but, as the velocity of the moving parts approaches its maximum in the neighborhood of mid-stroke, the effect of inertia is to lessen the force exerted by the cord and therefore its stretch. This continues throughout the forward stroke; as the length of the cord increases its stress decreases, and the net effect is to maintain a reasonably uniform total stretch and consequently to give a correct drum motion. On the return stroke, however, the spring tension is required to accelerate the wheel sufficiently so that the cord does not slacken, and it must do this not only against inertia, but against the frictional resistance.

As the friction is large, the spring may not be able to do this at a rate equal to that at which the cross-head is being accelerated, particularly at the beginning of the stroke, as here the acceleration is greatest. The stretch of the cord is therefore released at the beginning of the return stroke, if not entirely slackened, as evidenced by whipping. Beyond mid-stroke, the moving parts of the wheel have gained sufficient momentum to draw up the cord again, and this momentum keeps it taut to the end of the return stroke. Under ordinary circumstances, then, a marked distortion of the drum motion may be expected only through the return stroke and greatest at its beginning.

Obviously the length of the cord free to stretch, i.e., that part of it not wrapped around the wheel, has a decided influence upon the correctness of the motion. In practice this length at its minimum and maximum is determined by the engine stroke. Ordinarily, at the head end, the length of cord free to stretch is about equal to the stroke, and at the crank end twice as long.

Experiments upon reducing wheels will disclose the following characteristics.

First. The overtravel of the indicator drum is small at both ends of the stroke.

Second. The distortion of the return-stroke motion is very marked and follows a characteristic wave. The forward motion is in all cases very closely true.

Third. The piston speeds producing whipping and therefore limiting the operation of the wheel, are much greater at the longer strokes.

The first effect is largely due to friction which helps the springs to check the velocity of the wheel at the end of the forward stroke. Much overtravel at the crank end would be evidenced by whipping of the cord. For instance, if the drum motion were 4 ins. and the engine stroke 24 ins., an overtravel of the drum of $\frac{1}{8}$ in. would mean a slackening of the cord six times that amount, or $\frac{3}{4}$ in., since any distortion of the drum motion is multiplied at the wheel by the ratio of the motions. This is enough to cause whipping unless the cord is very stretchable. At the head end the overtravel is subdued not only by friction but by the positive resistance of the cord, the stretch of which is a minimum at this end because of its reduced length.

Overtravel of the drum motion does not indicate error with reducing wheels, as it is always small, exists for even low speeds when the motion is actually accurate, and may not exist at all when the friction is such as to cause large error.

The second effect is due not only to friction, but is increased because an indicator cord varies very considerably in length for changes of stress below 1 lb., although above that amount the stretch is comparatively small and uniform with increasing stress. Also the cord will contract markedly when the stress reduces to zero, and sometimes has been observed to continue contracting after the stress was entirely removed. If the frictional resistance to motion of the reducing wheel prevents the spring from properly accelerating it upon the return stroke, the cross-head motion gains upon the wheel motion, thus reduc-

ing the stress in the cord, perhaps to only a few ounces. The cord accommodates itself to this change in length because it contracts markedly at low stresses, and not until the distortion of the drum motion is quite pronounced is the speed of contraction exceeded by the cross-head speed, as evidenced by whipping.

The lag of the drum motion just after passing the crank-end dead center is only momentary at moderate speeds because the force required to accelerate the wheel is a maximum at dead center; when it has once started upon the return stroke less force is needed to continue the motion and this the spring is able to deliver. At higher speeds the action of the spring is further delayed, so that the late acquired momentum of the parts causes them to overshoot the mark and fetch up against the positive resistance of the cord, resulting in the wave effect.

The third effect follows from the limitations caused by inertia. It may be explained by the fact that at a given piston speed the oscillations per minute of the wheel are greater the smaller the engine stroke, and therefore the effect of inertia is greater with the small strokes. Obviously, it is more difficult to oscillate a mass moving at a fixed average speed, the more frequent are the oscillations. This also follows from the mathematical value for the force required to accelerate a reducing wheel at the ends of the stroke (assuming harmonic motion) which may be expressed as follows:

$$A = W \times L \times N^2 \times \text{a constant}$$

in which

A = Force of acceleration referred to the cord;
W = Weight of moving parts referred to the cord;
L = Length of engine stroke;
N = Revolutions per minute.

In this expression LN is proportional to the piston speed. The force A cannot be greater than a certain value which limits the

operation of the wheel. The product LN will have a maximum value, corresponding to the limiting value of A, by increasing L rather than N, since A varies as the first power of L, and as the square of N.

This analysis does not consider the inertia of the indicator drum. The throw of the drum is practically constant at all strokes, so the force required to accelerate it at the ends of the stroke is

$$a = w \times l \times N^2 \times \text{a constant}$$

or

$$a = w \times N^2 \times \text{a constant}$$

in which w is the weight of the drum and l its stroke. Thus, a varies with the square of the R.p.m. only.

The total force of acceleration of drum and wheel is the sum of these forces a and A, but as the drum spring is strong enough to meet the drum inertia at the highest speed of oscillation possible with the usual reducing wheel, the force a generally need not be considered.

If the wheel velocities are low so that inertia is not material, the frictional resistance to motion, if excessive, may cause a material lag of the drum behind its correct position throughout both strokes, since it effects a stretch of the cord upon the forward and a slackening upon the return stroke.

If the variation of stress in the cord is the same for different engine strokes, the distortion of the drum motion is the same. This is because any error in the motion of the wheel is reduced at the drum in the ratio of the drum travel to the wheel travel. For example, if the stresses are the same, the stretch of the cord on a 4-ft. engine stroke is double that of a 2-ft. stroke, since the cord is twice as long, but as the resulting drum distortion is $\frac{1}{12}$ of the stretch in the one case (drum stroke being equal to 4 ins.) and $\frac{1}{6}$ in the other, the effect is the same.

(a) Determination of Spring Tension and Friction. The method is the same as described under Test 10 (*a*) and (*b*). The friction in a well designed wheel is not likely to be less than 20 per cent of the spring tension, and may be as much as 50 per cent in poor designs or when the reducing wheel is badly assembled. As the greatest source of error is in friction, this test is an important one and furnishes a good indication of the merit of a reducing wheel.

When the wheel spring tension cannot be made great enough to accelerate the parts correctly, the deficiency may be made good by tightening the indicator drum spring, to a limited extent only. The drum spring acts upon the wheel at a mechanical disadvantage on account of the reduction of motion, and therefore has only a small effect to accelerate it. On the other hand, tightening the drum spring increases the reaction on the drum spindle, and therefore the friction to be overcome throughout the stroke, and this may cause even more distortion. As a general rule, it is best that the indicator drum spring have only just sufficient tension to overcome its own inertia. The wheel spring tension should be as high as possible if its ability to accelerate the wheel parts is doubted.

(b) Testing Reducing Wheels with the Drum Motion Tester. To adapt this device, Fig. 38, to reducing wheels, a bracket, or some equivalent attachment is necessary with a long stroke to duplicate an engine cross-head motion; and, to make the apparatus cover all operating conditions, the stroke should admit variation. This is conveniently accomplished by coupling an extension rod to the motion tester, carrying at its outer end a rack meshing with a pinion on a vertical shaft. This shaft *S*, Fig. 45, carries a wooden disc replaceable by others of different diameters. When the rack is reciprocated, the disc is given an oscillating motion. An extension of the indicator cord from the reducing wheel being fastened to a point on the circumference of the disc, the cord will be reciprocated through a stroke

the length of which depends upon the diameter of the disc. By using different discs, a reducing wheel may thus be operated through any desired stroke.

The reducing wheel to be tested is connected to an indicator and the whole set in place on the tester. The wheel may then be operated at any required conditions of stroke, speed, and cord length; and error diagrams similar to Fig. 39 taken. The cord length may be fixed by tying the desired length to a piece of stranded wire the other end of which is attached to the disc

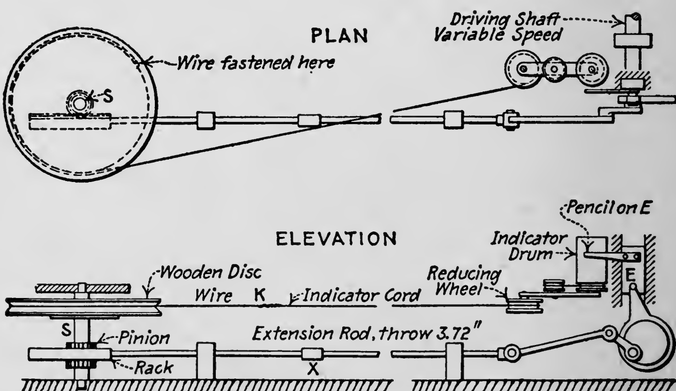


Fig. 45.—Smallwood's Drum Motion Tester Applied to Reducing Wheels.

of the tester. The wire being practically inextensible, the effect is to reciprocate the cord through the desired lengths as by an engine cross-head at K, Fig. 45.

Fig. 46 shows a number of error diagrams taken in this way.

(c) Correction of Indicator Diagrams. This may be done as described under Test 11 (*f*). The reducing wheel is reasonably accurate on short engine strokes and moderate piston speeds, and on long engine strokes at all usual piston speeds *if properly designed to avoid friction.* The drum motion resulting from its

use has its maximum error, when the friction is small, at a part of the stroke where usually it would affect the indicator diagram little if any, namely, at the beginning of the return stroke where either the steam line on the crank end or the exhaust line

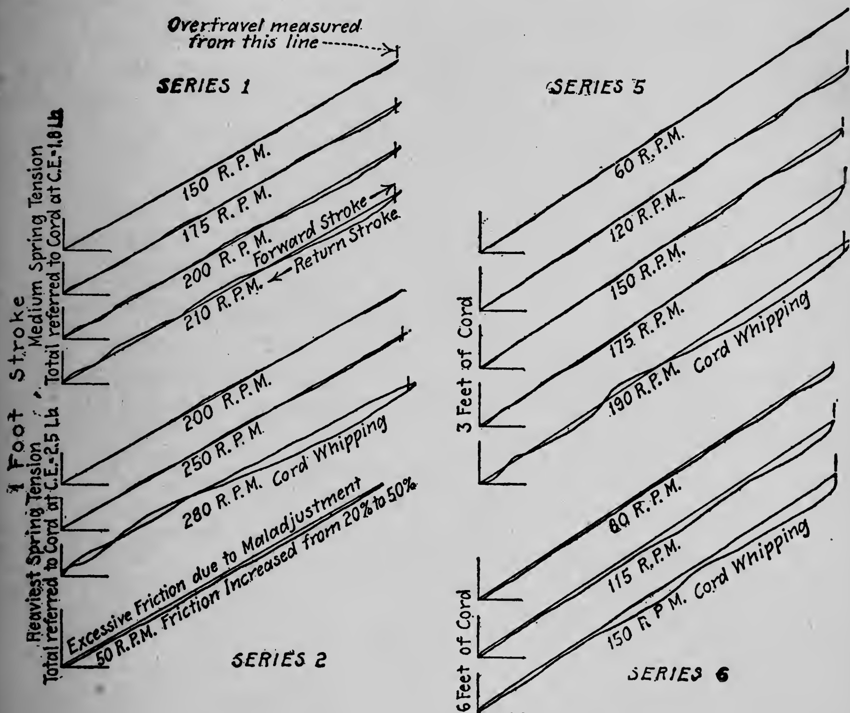


FIG. 46.—Error Diagrams from Reducing Wheels.

on the head end diagram is described. As these lines are generally horizontal, or nearly so, errors upon them would not appear, except at their ends.

IRREGULAR AREAS AND MEAN HEIGHTS

Numerous engineering instruments, of which the engine indicator is one, have been devised to give an autographic diagram of the measured quantity expressed as an ordinate, the abscissæ being generally time or in linear units simply. From such a diagram it is often desired to get the mean value of the measured quantity, that is, the mean ordinate of the diagram to scale. For this purpose, *planimeters* are used. Some planimeters measure the mean ordinate directly; others measure the area of the diagram, from which the mean ordinate may be found by dividing by the length. The best known of these two types are the Amsler Polar planimeter and the Coffin averaging instrument. The former is used to measure any irregular area; the latter is applied chiefly to the determination of the mean height of indicator diagrams.

14. The Polar Planimeter

Principles. Fig. 47 shows in diagram the Amsler planimeter. When the tracing point traverses any closed curve, 1–2–1 the record wheel follows in a certain path and, through contact with the surface upon which it bears, is given a motion partly rolling and partly sliding. The principle upon which the instrument depends is that the rolling motion of the wheel is directly proportional to the area circumscribed by the tracing point. If, then, the wheel is properly graduated, the area may be read directly.

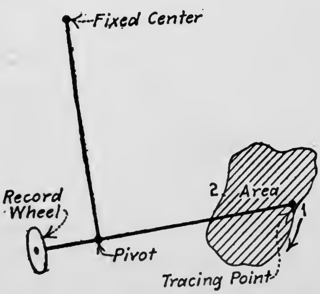


Fig. 47.—Polar Planimeter.

When the area to be measured is comparatively large, the

fixed center of the instrument is placed within the area, thus securing a greater reach of the tracing point. (See Fig. 51.) In this case, the readings of the wheel must be interpreted differently and the constant known as *the zero circle* must be known. The zero circle may be defined as one of such size that if the fixed center of the planimeter is placed at its center, and if the tracing point then traverses its circumference, there will be no motion of the record wheel. It may be seen from Fig. 48 that this condition requires that the plane of the record wheel shall pass through the fixed center, for in this position the wheel will travel in the direction of its own axis and there can therefore be no rolling.

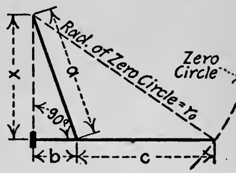


FIG. 48.

The algebraic expression for the radius of the zero circle may be deduced as follows. (See Fig. 48.)

$$\begin{aligned} r_0^2 &= x^2 \qquad + (b+c)^2 \\ &= (a^2-b^2)+(b^2+2bc+c^2) \\ &= a^2+c^2+2bc \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1) \end{aligned}$$

To deduce the mathematical relation between the area circumscribed by the tracing point and the circumferential motion of the record wheel. There are two cases, namely, the fixed center outside and the fixed center inside the area circumscribed. The following paragraph gives an outline of the deduction for the first case.

1. The motion of the tracing point is referred to polar coordinates whose pole is at the fixed center of the instrument. A polar differential of area is considered as the one circumscribed by the tracing point. An algebraic expression for this area is obtained in terms of the polar coordinates. 2. An expression is deduced for the motion of a point on the record wheel circumference in

terms of the constants of the instrument. The motion is that which takes place when the tracing point has outlined the differential of area. 3. By comparing the two expressions (for the area and for the motion of the wheel) the desired relationis obtained.

Fig. 49 shows the differential of area, marked 4–5–6–7. Using the notation of the figure,

$$\begin{array}{lll} & \text{Area } 1\text{–}4\text{–}5 & = \tfrac{1}{2}r\cdot rdK = \tfrac{1}{2}r^2dK \\ & \text{Area } 1\text{–}7\text{–}6 & = \qquad\qquad \tfrac{1}{2}r_1{}^2dK \\ \hline \text{Subtracting,} & \text{Area } 4\text{–}5\text{–}6\text{–}7 & = \qquad\qquad \tfrac{1}{2}dK(r^2-r_1{}^2). \end{array} \qquad (2)$$

This is the expression for the differential of area.

Consider now the corresponding motion of the record wheel. It is to be observed first that this is not affected by the radial motion of the tracing point. The angle between the arms when the point is at 4 (Fig. 49) is the same as that at 5. The same applies to points 6 and 7. Hence, the motion of the wheel when the tracing point passes on the radial from 5 to 6 is the same as that when it passes from 7 to 4. But as these two motions are opposite, they neutralize each other. The same reasoning applies to any irregular area as in Fig. 47. The radial component of the motion from 1 to 2 is the same in amount as that caused in passing from 2 to 1, but opposite in direction. So we may altogether disregard the motion of the wheel produced by the radial motion of the point.

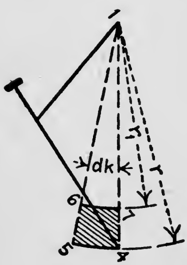


FIG. 49.

Considering again the differential of area of Fig. 49, it is seen that the motion of the tracing point from 4 to 5 is greater than that from 6 to 7, and that the angle made by the arms is different when the point traverses the two arcs. Hence the circular component of the motion of the tracing point when traversing a closed curve produces a record on the wheel. We have, then, to consider the effect of this component.

See Fig. 50. The record wheel moves from 2 to 2′ when the tracing point passes from 4 to 5. The rectangular component of the motion, 2–2′, causing rotation of the wheel, is represented by the line m, perpendicular to the wheel axis. m is therefore

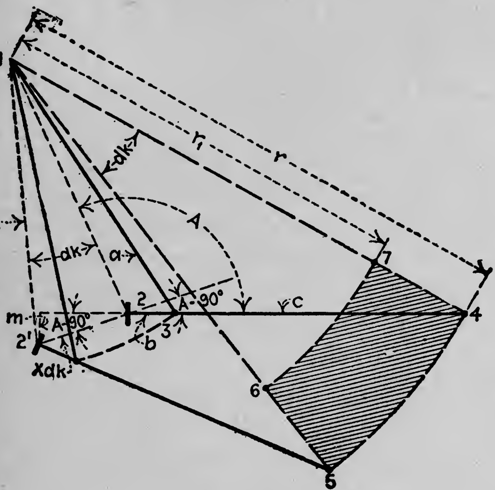


Fig. 50.

the distance moved through by any point on the record wheel circumference relative to its axis.

$$m = XdK \sin (A - 90°)$$
$$= -X \cos A dK \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

From 1–2–3, $\quad a^2 = b^2 + X^2 - 2bX \cos A.$

From 1–2–4, $\quad r^2 = (b+c)^2 + X^2 - 2(b+c)X \cos A.$

Subtracting, $\quad a^2 - r^2 = -2bc - c^2 + 2cX \cos A.$

From which, $\quad X \cos A = \dfrac{1}{2c}(a^2 + c^2 + 2bc - r^2).$

From (1), $\quad = \dfrac{1}{2c}({r_0}^2 - r^2) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$

Combining (3) and (4)

$$m=\frac{dK}{2c}(r^2-r_0^2), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

which is the value for the rotation of a point on the circumference of the record wheel when the tracing point moves from 4 to 5. Similarly, when it moves from 6 to 7, the motion is

$$m_1=\frac{dK}{2c}(r_1^2-r_0^2).$$

The difference between these two motions, m and m_1, is the resultant motion M of a point on the circumference of the record wheel when the whole area has been circumscribed, or

$$M=m-m_1=\frac{dK\,(r^2-r_1^2)}{2c}. \quad . \quad . \quad . \quad . \quad . \quad (6)$$

Comparing (2) and (6)

$$\text{Area 4–5–6–7}=\frac{dK}{2}(r^2-r_1^2)\cdot\frac{c}{c}$$

$$=Mc, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

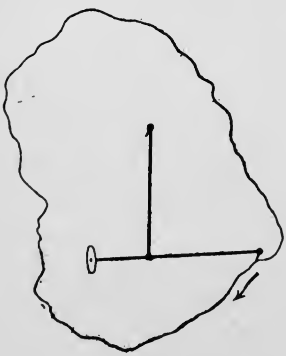


Fig. 51.

which is the required relation.

The deduction of the relation for the second case, namely, when the fixed center is within the area to be evaluated, is similar. Under this condition, however, the circular component of the tracing point is always in one direction. Hence there is no subtraction as indicated by equation (6) and the differential expression is

$$M=\frac{dK}{2c}(r^2-r_0^2).$$

Integrating this for the entire circumferential motion of a point on the circumference of the record wheel,

$$M=\int_0^{2\pi}\frac{dK\,r^2}{2c}-\int_0^{2\pi}\frac{r_0{}^2dK}{2c}=\int_0^{2\pi}\frac{d(\text{area})}{c}-\int_0^{2\pi}\frac{r_0{}^2dK}{2c}$$

$$=\frac{\text{area}}{c}-\frac{2\pi r_0{}^2}{2c}.$$

From which, $$\text{Area}=Mc+\pi r_0{}^2, \qquad (8)$$

which is the required relation.

From these two deductions it is seen that,

First. When the fixed center of the planimeter is outside the area to be integrated the motion of a point in inches, on the circumference of the record wheel, relative to its axis, multiplied by the length of the arm c, equals the desired area.

Second. When the fixed center of the planimeter is inside the area to be integrated, this product added to the area of the zero circle equals the desired area. With the working form of the instrument, the multiplication Mc is not necessary, the wheel being graduated in terms of square inches, or other units of area.

If N is the number of turns of the record wheel on its axis, and w the diameter of the record wheel in inches, these relations may be expressed as follows:

First. $$\text{Area}=N\times\pi w\times c. \qquad (9)$$

Second. $$\text{Area}=N\times\pi w\times c+\pi r_0{}^2. \qquad (10)$$

In some types of polar planimeter, the arm c is made adjustable in length, so that areas drawn to various scales may be read directly.

As an exercise, the student should set the arm c at any random length, and then by traversing a known area and noting the number of turns N, show that equation (9) holds true.

(a) **Determination of the Zero Circle.** If the lengths of the arms, a, b, and c, are carefully measured, the radius of the

zero circle may be computed by means of the relation given by equation (1), and from this the area.

If the axis of the wheel is in a perpendicular plane through the arm c, the graphic method suggested by Fig. 48 may be used. Two lines are drawn at right angles to each other, the fixed center placed on one, the tracing point on the other, and the point of contact of the wheel and the paper at the intersection of the lines. The radius of the zero circle is then the distance between the fixed center and the tracing point. If the axis of the wheel does not lie in a vertical plane through the arm c (as is sometimes the case) the construction must be altered to allow for this difference.

When the value r_0 is found, it may be checked by noting a zero motion of the wheel when the tracing point moves as in Fig. 48. The arms may be clamped by placing the fixed and tracing points in two holes pierced in a strip of manilla paper, these holes being distant from each other an amount equal to the radius of the traversed circle.

(b) Comparison of Instrument Indications with Known Areas. For this purpose may be used a check rule, an instrument by which the tracing point of the planimeter may be swung through a circle of known radius. Initial and final readings of the record wheel are taken; the difference between these readings should equal the area traversed. When the fixed center of the planimeter is inside the area, the difference between the readings should be added to the area of the zero circle.

If the instrument indications do not correspond to the actual areas, the length of the arm, c, should be adjusted.

Notice that the wheel reads positively if the direction of the tracing point is clockwise, with the single exception of the case when the area to be integrated is less than the zero circle and the fixed center is inside the area.

It is well always to traverse the area in a clockwise direction and to use the following forms:

$$\text{Area} = \text{second reading} - \text{first reading},$$

or

$$\text{Area} = \text{second reading} - \text{first reading} + \pi r_0^2.$$

As an exercise record the results of two or three area readings of the following and compare with their calculated values.

Small circle, fixed center outside.

Circle > zero circle, fixed center inside.

Circle < zero circle, fixed center inside.

Note Carefully. Do not put down the result of an area only, but record in tabular form, as follows:

1st Reading.	2d Reading.	Result.

(c) **Comparison of Instrument Indications with Areas to Scale.** Planimeters with adjustable arms generally are arranged to be applicable to various scales. For instance, for a certain adjustment, each graduation of the wheel may indicate one square foot on an area drawn to a linear scale of $\frac{1}{4}$ in. equals 1 ft. For comparison the check rule may be used, its area in square feet on a $\frac{1}{4}$-in. scale being figured by multiplying its area in square inches by the square of the linear scale, that is, 16.

As an exercise, compare the planimeter and calculated results of a circle larger than the zero circle, using the scale setting.

(d) **Arm Adjustment to a Required Scale.** Suppose it is required to read areas to a scale not provided for by the markings of the adjustable arm. The necessary length of the arm c may be calculated from equation (7). In this equation, we may assume the area to be one square inch. Then the motion M equals the length of one graduation on the wheel multiplied by the number of graduations previously selected to represent the scale units in one square inch.

Let w = diameter of record wheel, inches.
G = number of graduations on the record wheel.
X = number of graduations representing one scale unit of area.
Y = number of scale units of area in one square inch.

Then, length of one graduation $=\frac{\pi w}{G}$ in inches.

When the tracing point traverses one square inch of area, the number of graduations corresponding to the motion, M, will be XY; and

$$M=\frac{\pi w}{G}XY.$$

From (7) $$\frac{\pi w}{G}XYc=1 \text{ sq. in.},$$

and $$c=\frac{G}{\pi wXY},$$

in which everything is known except c, which may be found.

A convenient application of this principle is in the direct determination of horse-power from indicator diagrams, for which purpose the arm c should be adjusted to the length given by the following formula.

$$c=\frac{10500lG}{SKwX}, \qquad (11)$$

in which G, w, and X are as previously defined, and
l = length of the indicator diagram, inches,
S = scale of the indicator spring,
K = product of LAN in the $PLAN$ formula (see Test 43).

Problem 14_1. Deduce the equation for the zero circle of a planimeter having the record wheel between the pivot and the tracing point instead of as shown by Fig. 48.

Problem 14_2. If a is 5 ins.; b, 2 ins.; and c, 3 ins., what is the area of the zero circle applicable to an arm adjustment for a $\frac{1}{4}$ in. = 1 ft. scale?

Ans., 2310 sq. ft.

Problem 14_3. The fixed center of a planimeter is inside an area to be integrated which is smaller than the zero circle; consequently the wheel rotates backward. The first reading is 29.23 sq. ins.; the second, 48.73 sq. ins. What is the area if the zero circle area is 212 sq. ins.?

Ans. 131.5 sq. ins.

Problem 14_4. Integrate the area enclosed by a loop like a figure 8, first by finding separately the areas of the loop, and second by tracing the loop in the direction of its curve. Account for differences.

Problem 14_5. The arm c of a planimeter is 2.0 ins. long and the diameter of its record wheel is 0.79 in. How many turns will the wheel make when the tracing point circumscribes 117 sq. ft. to a $\frac{3}{4}$-in. = 1 ft. scale?

Ans. 13.1.

Problem 14_6. If the wheel diameter is 0.79 in., what should be the length of the arm c so that one revolution corresponds to 1000 square miles on a linear scale of $\frac{1}{8}$ in. = 1 mile? *Ans.*, 6.25 ins.

Problem 14_7. Deduce equation **(11)**.

Problem 14_8. Find the mean height of a given indicator diagram with a planimeter. Calculate the horse-power. Find the horse-power by adjusting the arm as described under (d). Compare results.

15. The Coffin Planimeter

Principles. Fig. 52 represents this planimeter. It consists of a single arm one end of which carries the tracing point, the other end bearing on a pin which slides in a guide represented by the line gg. The record wheel is mounted on the arm as shown. An examination of Figs. 52 and 47 will show that the Coffin planimeter is the same in principle as the polar; the mechanical difference being that the arm a of the former is infinitely long, so that the pivot swings in a right line instead of the arc of a circle. The relation Mc = area therefore applies (see Test 14, principles).

The instrument may be used for finding areas, but its chief use is in determining the mean heights of indicator diagrams.

For this purpose the indicator diagram is arranged as shown by Fig. 53, with its left-hand extremity in the line of the guide, and its base perpendicular to the line of the guide. The tracing point of the planimeter is then placed at the extreme right-hand point of the diagram, 1, and an initial reading taken. Next, the figure is traced in a clockwise direction, until the tracing point comes back to the point, 1. If now the tracing point is moved on a line parallel to the guide line until the wheel indicates again its initial reading, the distance 1–2 thus traveled by the tracing point equals the mean height of the indicator diagram in inches.

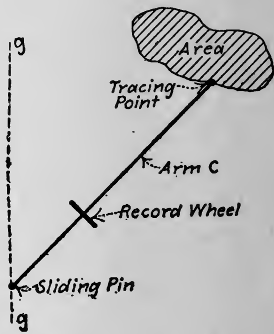


FIG. 52.
Coffin Planimeter.

To prove this, it should first be mentioned that the motion of the wheel corresponding to the motion of the point from 1 to 2 is the same as that corresponding to the motion of the point around the area, since the wheel returns backward to its initial position during the motion 1–2.

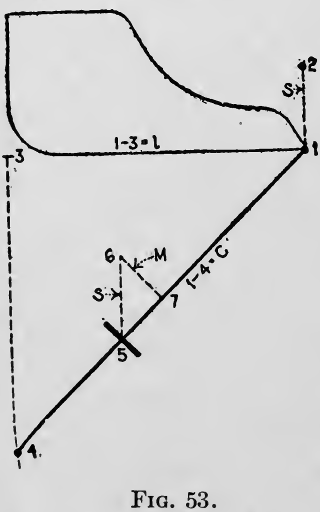


FIG. 53.

In Fig. 53, the motion 1–2 of the point causes a change of position of the wheel from 5 to 6. The line 6–7 is the component of this motion which causes rolling of the wheel. Since this rolling is the same as that taking place when the area is circumscribed, it equals M in the formula $Mc =$ area, the area being that of the indicator

diagram, and c the length of the planimeter arm (see Test 14, principles). Also,

$$\text{area} = h \times l$$

in which h is the mean height and l the length of the diagram. Hence

$$Mc = hl, \text{ from which } h = \frac{Mc}{l}.$$

From the similarity of triangles 1–3–4 and 6–7–5,

$$\frac{S}{M} = \frac{c}{l}, \text{ from which } S = \frac{Mc}{l}.$$

Hence $S = h$, that is the distance 1–2 equals the mean height of the indicator diagram.

(a) Comparison of Records with Known Mean Heights. Instead of an indicator diagram, a carefully laid out rectangle may be used. The tracing point should be started in the lower right-hand corner, and the figure traced back to this starting point. The parallel motion necessary to bring the wheel back to its initial reading will take the point back to the upper right-hand corner of the rectangle, if the instrument indicates correctly. This test should be repeated several times, care being taken that the point pursues the path of the rectangle closely.

If the instrument does not indicate correctly, it may be because the length of the arm has been altered, through bending of the tracing point or otherwise, because of faulty graduations of the wheel, or because of the wheel sticking instead of revolving freely.

Problem 15_1. If the length of the arm c is 5 ins., what should be the diameter of the wheel so that one revolution corresponds to 10 sq. ins. of area?

Ans., 0.636 in.

Problem 15_2. If the test under (a) gives results uniformly 8 per cent too small, is the arm too long or too short, and how much should it be changed if its length is 6.12 ins.?

Ans., 0.45 in.

Problem 15_3. Using the indicator diagram of Problem 14_8, find the mean effective pressures by the Coffin planimeter, and compare results with those from the Amsler.

16. The Averaging of Circular Charts

Principles. The growing use of autographic instruments of precision yielding circular charts has led to the development of appropriate methods of averaging and integrating. Recording instruments are used both for the measurement of quantities which need not be totaled, such as pressures, temperatures, CO_2 percentages; and for time rates, such as cubic feet of steam or pounds of water per minute. It is often desirable to average such records, and it is essential, in the case of time-rate ones, to get total quantities. The latter may always be had, when the *average* rate is known, by multiplying this rate by the time. The problem, then, resolves itself into one of finding averages.

It should be noted, however, that there are available special planimeters yielding total quantities from time rate charts, and that many recorders are equipped with automatic ones, so arranged as to be driven by the same mechanism that moves the chart. Averaging methods only will be considered here.

Circular diagrams always have uniform angular coordinates, since they are obtained by clockwork moving in proportion to time. The radial coordinates, on the other hand, may be uniform or non-uniform, depending upon the law controlling the pen movement and its mechanism. When the ordinates are non-uniform, the usual methods of averaging do not apply.

(a) **The Radial Planimeter.** Fig. 54 illustrates the principle of the Bristol-Durand instrument for averaging circular charts. This consists of an arm carrying a record wheel and tracing point at one end, arranged to slide in a pivoted sleeve. With the pivot set at the center of a circular chart, the tracing point can traverse any curve on the chart, the arm sliding in the sleeve when the point moves radially. Since the record wheel axis is set on a radial line, the motion of a point on the wheel circumference with relation to its axis will be due only to circumferential motion of the tracing point. Radial motion of the

tracing point will cause no motion of the wheel on its axis. Thus it is seen that it is only the circumferential component of the motion of the tracing point which causes rolling of the wheel, and since the circumferential component is proportional to the radius, it follows that the rolling will be proportional to the radius.

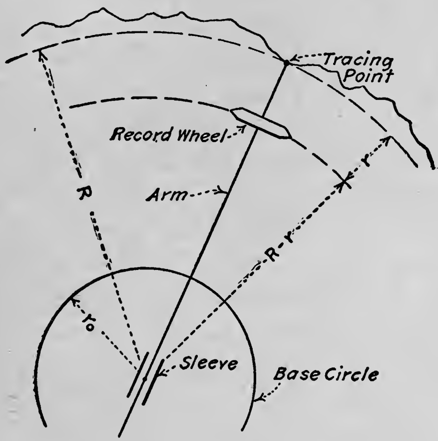


FIG. 54.—Radial Planimeter.

To deduce the equation of the instrument, let

R = average distance of tracing point to pivot center = average radius of curve, inches;

r = distance between tracing point and plane of wheel, inches;

r_0 = distance between center of chart and zero of circular coordinates, inches;

d = diameter of wheel, inches;

n = number of turns of wheel when tracing point traverses a given curve;

f = fraction of the time included between the curve limits to the time represented by a complete revolution of chart;

Then the motion of a point on the wheel circumference will be

$$\pi dn = 2\pi(R-r)\times f,$$

the quantity on the right being the circumferential component of the wheel motion which exclusively produces rolling. Transposing

$$R-r=\frac{\pi dn}{2\pi f}=\frac{dn}{2f}.$$

If r is made equal to r_0 (adjustment for this is provided), then the value of $R-r$ above is the average radius of the curve in inches, measured from the base circle. Multiplying by the scale of radial coordinates gives the desired mean.

If the tracing point is *between* the wheel and the pivot, the equation becomes

$$R+r=\frac{dn}{2f}.$$

To get the mean height from the base circle, in this case, it is necessary to multiply the number of turns, n, by $d/2f$ and subtract from their product r and r_0.

Calibration curves can be made for ready use with charts of given form.

When the radial coordinates are curved (that is, produced by a pen swing around a center), an error will result if the ends of the curve do not join. To obviate this, the tracing point must be brought back to the same radial distance from the chart center, at the finish as that at the start; and this closing must be made by following a curved radial coordinate.

The instrument may be tested for accuracy, first, by tracing a circle of known radius; and, second, by noting whether a wholly radial movement of the tracing point rolls the wheel.

(b) Approximate Average with Integrating Planimeter. This instrument is not appropriate to circular diagrams as might at first appear. If the area of such a diagram be found with the

polar planimeter and divided by its angular measure expressed in radians, and the square root of this quotient taken, the result will be the square root of the mean of the squares of the radii, instead of the arithmetical mean. However, in cases where the record is not that of a very variable quantity, these two values are not materially different, so that the one can be taken in lieu of the other. The procedure is as follows:

Close the gap at the ends of the curve to be averaged by radial lines, and then find the area bounded by the curve and the radials. Then, if f is the fraction of the time included between the radials to that represented by a complete chart revolution, the mean height of the curve from the center of the chart is (approximately)

$$R=\sqrt{\frac{\text{Area}}{\pi f}}.$$

The average quantity in scale units is now found by noting the value of a division on the chart at a distance from the center equal to this mean height.

FLUID VELOCITIES—Meters

Meters for the measurement of fluid quantities are of two broad types: those that measure volume or weights directly and those that measure velocities.

Volume meters are so arranged that all of the measured fluid passes through them, alternately filling and emptying compartments and thereby displacing a moving part which registers through a gear and counter combination, the quantity passed. This type of meter gives the total quantity at any time. To get rates it is necessary separately to count the time.

Velocity meters measure quantity by means of the relation that the flow in volume units per unit of time is equal to linear velocity multiplied by the cross-sectional area of the fluid. Such

meters are dependent upon the uniformity of velocity throughout the cross-section, or upon the correctness of an estimated average velocity when it is not uniform. They may be calibrated to give volumes or weights per unit of time. To get total quantities, it is necessary to multiply by the time.

17. Calibration of a Volume Water Meter

Principles. In some types of water meter, the moving part is a piston or disc which is displaced by the water entering the compartment of which this moving part is a wall. There may be leakage past the moving part or the valve motion which controls it. This leakage will increase with the pressure drop through the meter. The greater the pressure urging the water through the meter, the faster is the flow. Hence, the accuracy of a water meter will vary with the rate of flow. Other types of water meter, which may be arranged to avoid such leakage, still will have variable accuracy owing to the effect of inertia, etc., at different rates. For a complete calibration, then, a meter should be tested at enough different rates to cover its range.

(a) **Calibration against a Calibrated Tank or Scales.** The meter is arranged so that the water passing through it can be weighed in a tank placed on a platform scales or measured in volume from the known dimensions of the tank. If the volume is measured and the instrument reads in pounds, or vice versa, it is necessary to know the density of the water with reference to its temperature. Enough readings of both the true quantity as shown by the scales or tank and of the meter should be taken to get fair values of the rates of flow. In this connection, rules 4 and 5 given on page 8, should be carefully followed. From the observations, should be figured a series of " true rates " in pounds, gallons, or cubic feet per unit of time, and of " rates by the meter " in the same units. If desired, these may be plotted for a calibration curve.

(b) Curve of Correction Factors. This is more convenient to use with a volume meter than a calibration curve, because the instrument is not as a rule used to get rates, but total quantities. The correction factor at any rate of flow is that number by which the total quantity as shown by the meter for any length of time is multiplied to get the true quantity. Consequently, the correction factor is the ratio of the true rate to that shown by the meter. A series of values of the factor may be calculated from the calibration curve and plotted against the rate by the meter.

In using the curve, to select the appropriate correction factor, a rough value of the rate by the meter is figured, and from the curve the corresponding factor is obtained. The true quantity for the total time is then readily determined.

Problem 17_1. Discuss the following observations taken from the test of a meter, and from them figure the true rate, the rate by the meter, and the correction factor.

Time.	True Weight.	Weight by Meter.
2:10	50.5 lbs.	10.0 lbs.
2:15	68.8	20.8
2:20	80.1	31.5
2:25	91.3	42.1

The first true weight is that of an empty tank on a platform scales.

Ans., 2.25 and 2.13 lbs. per min.; 1.06.

Problem 17_2. If a water meter of the volume or displacement type is accurate at all rates when the water is 60° F., draw its calibration and correction factor curves to be used when the water is at 120° F. (See p. 369.)

18. Calibration of a Volume Gas Meter

Principles. The gasometer is the most accurate instrument of this type. It is represented by Fig. 55, and consists of two tanks as shown, the upper one being movable vertically and properly counterbalanced. This arrangement makes a chamber of variable size and water sealed. When the upper tank is raised, gas is drawn through the inlet pipe, the valve *O* being closed and *I* open. When lowered, the gas is discharged, the valve control

being reversed. The volume of gas thus displaced is measured by the vertical motion of the upper tank, its cross-section being known.

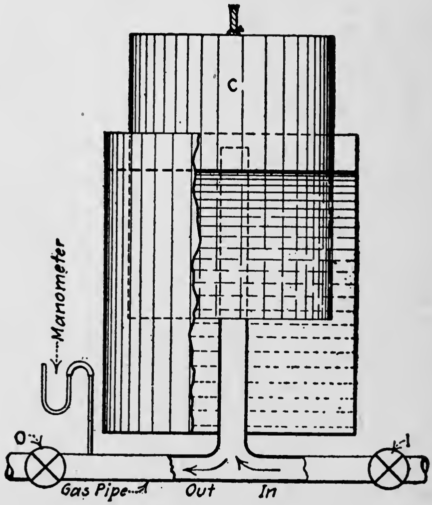


Fig. 55.—Gasometer.

It will be noted that the gasometer cannot be used for measureing continuous flow unless two are operated, one to fill and one to discharge, alternately.

Another form of gasometer is shown by Fig. 56. Gas is drawn into the cylindrical chamber C by allowing water to flow out through the outlet pipe. This gas is then displaced through the gas outlet by causing water to enter through the water inlet, the valves being properly adjusted. The gas displaced is measured by the rise of water level in the gage glass.

Commercial forms of gas meter are generally of the "dry meter" type and are arranged to measure continuous flow. In this type there are two bellows chambers which are alternately filled and emptied. One side of each bellows being stationary,

the other one is thus given a motion which actuates through a linkage the valve control and the recording mechanism.

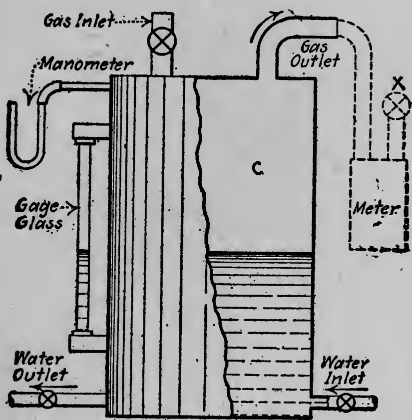


Fig. 56.—Gasometer.

As gas meters in general record volumes under the existing conditions of pressure and temperature, these conditions should be noted if it is desired to translate the readings into weights or into volumes under standard conditions. (See p. 171.)

(a) **Calibration against a Gasometer.** The meter to be tested is arranged as shown by the dotted lines of Fig. 56. The gas used for testing may be air; the gas inlet may then draw through an open pipe. The gasometer being full of air, water is caused to enter through the water inlet, its rate of flow being the desired gas rate. This is obtained by manipulating the water inlet and noting the time of rising in the gage glass. The valve in the outlet pipe at the meter is then adjusted so as to give a constant air pressure as shown by the manometer which, if equal to that usual in city gas mains, should be about four inches of water.

For other details, see Test 17 (*a*).

(**b**) **Curve of Correction Factors** is obtained the same as for a water meter, Test 17 (*b*).

Problem 18_1. In calibrating a gas meter the capacity of which is 500 cu. ft. per hour, what rates should be applied, and how long should each trial be so that the error due to reading the gage glass is no more than 1 per cent? The diameter of the gasometer is 3 ft.

Problem 18_2. What should be the capacity of a gasometer to calibrate a meter of 2000 cu. ft. per hour capacity?

19. Calibration of a Weir

Principles. A weir is a water meter of the velocity type. It is formed by a notch made in a dam like obstruction in a stream of water through which notch all of the water is caused to flow. The level of the water behind the dam stands above the bottom edge or sill of the notch, and according to a definite difference in these levels, a definite velocity is attained by the water, and hence the quantity passed in a unit of time. If it is arranged that the difference of level be measured by a precise instrument, such as a hook gage, then the flow may be calculated; or, if the weir has been calibrated, the flow may be obtained from the calibration curve.

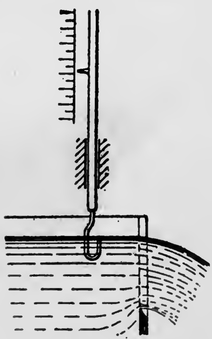


Fig. 57.
Weir and Hook Gage.

Fig. 57 shows a weir, the height of the water level above the sill being represented by H. The pressure to which the particles of water at the sill are subjected is equal to H ft. of water. When this pressure is reduced to zero by emergence of the water into the atmosphere, the work done is WH ft.-lbs. per second, W being the weight of water in pounds per second. This work is transformed into kinetic energy, the expression for which is $\frac{WV^2}{2g}$, V being the

velocity in feet per second, and g the acceleration of gravity. Consequently, $WH = WV^2/2g$ and,

$$V = \sqrt{2gH}.$$

The velocity of the particles of water at levels above the sill is less than this since they are at less pressure, and it may be shown that the average velocity is $\frac{2}{3}V$ or $\frac{2}{3} \times \sqrt{2gH}$. If the breadth of the weir is B ft., then the cross-sectional area to which this average velocity applies under ideal conditions is BH sq. ft. Then, since

$$\text{Quantity} = \text{area} \times \text{velocity},$$

it follows

$$Q' = BH \times \tfrac{2}{3}\sqrt{2gH}$$
$$= \tfrac{2}{3}B\sqrt{2gH^3},$$

Q' being in cubic feet per second.

This is for ideal conditions. Actually, the velocity is somewhat less than the ideal because of eddies and friction of which the theory takes no account. So we may write

$$\text{Actual velocity} = C_1 \times \text{ideal velocity}.$$

C_1 is less than unity and is called the "coefficient of velocity."

Also the actual water section is less than BH since there is generally a contraction due to a fall at the top and a narrowing at the edges of the weir on account of the tendency of the water at the sides to continue in the plane of the weir. Hence,

$$\text{Actual area} = C_2 \times BH$$

in which C_2 is the "coefficient of contraction."

It follows that the actual quantity

$$Q=\tfrac{2}{3}C_1C_2B\sqrt{2gH^3},$$

$$Q=\tfrac{2}{3}CB\sqrt{2gH^3},$$

C being called the " coefficient of discharge."

For rectangular weirs with sharp edges, C equals about 0.62, depending upon the size of the weir and the head.

When the stream behind the weir is small in cross-section, its velocity is relatively high. The so-called velocity of approach will then appreciably increase the velocity of the water through the weir since it acts in addition to gravity, and should be allowed for. The average velocity of approach may be calculated by multiplying the velocity through the weir by the ratio of water sections at the weir and in the flume behind the weir. The additional head urging the water through the weir is then

$$\frac{(\text{Velocity of approach})^2}{2g}$$

which may be added to H for use in the formula for Q. This is an approximation since the velocity in the flume is very variable throughout the cross-section. The actual velocity of approach is greater since the water in the middle of the stream, having greatest influence on the passage through the weir, is of higher velocity than at the sides and bottom. It is more correct, then, to multiply the velocity head as just given by a number greater than unity which, according to Hamilton Smith, lies between 1.0 and 1.5.

Many modified formulas have been proposed for weirs to allow for the variations of C_2 and C. The most notable, perhaps, is Francis'

$$Q=3.33\ (B-0.2H)\sqrt{H^3},$$

in which 3.33 equals $C\,\frac{2}{3}\sqrt{2g}$. In this, it is regarded that each end contraction increases with the head and equals $0.1H$. There is thus less variation in the applied value of C which remains approximately equal to 0.62. This formula may be used for uncalibrated weirs having B greater than 4 ft. when the head, H, exceeds 5 ins.; with less than 1 per cent of error.

Other shapes of weir notches are the V-notch and the trapezoidal. For the former the equation is

$$Q=C\tfrac{8}{15}\sqrt{2gH^5}$$

when the sides of the notch make a right angle. See Test 20.

There is less variation in the coefficient of contraction for this type of weir than the rectangular. Also it is useful for variable flow and small quantities since the head diminishes at a rapidly decreasing rate with the quantity.

Merriman recommends an average value of C, 0.592, which gives

$$Q=2.53\sqrt{H^5}.$$

V-notch weirs are now made with elaborate indicating and recording apparatus to register the flow. (See Test 20.)

The trapezoidal weir may be arranged so that the extra breadth due to the slope of the sides as the head rises balances the increased contraction of section so that the product of the breadth B at the sill and the coefficient of contraction remains nearly constant. The slope of the sides should then be 1 to 4. The value of C may be taken as 0.62, and we have, as for a rectangular weir without end contractions,

$$Q=3.33\ B\sqrt{H^3}.$$

The difference of level, H, is generally measured by the hook gage, Fig. 57, by which minute differences of level may be detected.

In use, the hook is started just below the water level, then raised slowly until it barely breaks the water surface, fastened in that position and then read at the graduated scale.

(a) **The Zero of the Hook Gage is** its reading at a given datum plane. For a weir, this plane is the horizontal one through the lowest edge of the notch. To get the zero of the hook gage for a weir, a straightedge should be placed with one end at the lowest point of the notch and the other end on the point of the hook, and so balanced that very little weight comes on the hook, to prevent springing it. The hook is then raised or lowered until a spirit-level placed on the straightedge shows the horizontal, and a reading taken. The straightedge and level are then turned end for end and the procedure repeated. By averaging the two readings, any untruth of the straightedge or level is eliminated.

(b) **To Calibrate a Weir,** a series of readings of head and corresponding quantity rates in cubic feet per minute or second should be obtained and plotted. To get the quantity rate, the water may be discharged into a calibrated tank, the rate of rising in which will give the volume per minute. When the weir is at the end of a flume of uniform horizontal cross-section, a convenient method is to start with it empty, and then adjust the incoming water to the desired rate of flow. The rate can be ascertained as the flume is filling by readings of the hook gage taken at equal increments of time. When the water level has reached its highest position, uniform flow through the weir being established, a final reading of the hook gage gives the head corresponding to the rate thus previously ascertained. With this method, it is especially important that the rate be constant toward the end of each run.

(c) **The Coefficient of Discharge** varies with the rate. For any rate, it may be found from the quantity formula, the values of H and Q being known.

Problem 19_1. Find the coefficient of discharge from the following experimental data obtained from a rectangular weir. Size of flume 18 ft. long

by 3 ft. wide. Rate of rise, 0.12 ft. in 5 sec. Head on the weir, 5 ins. Breadth of weir, 18 ins. *Ans.*, 0.6.

20. Calibration of a V-notch Recorder

Principles. This instrument, largely used for measuring boiler feed, is illustrated diagrammatically by Fig. 58. A float

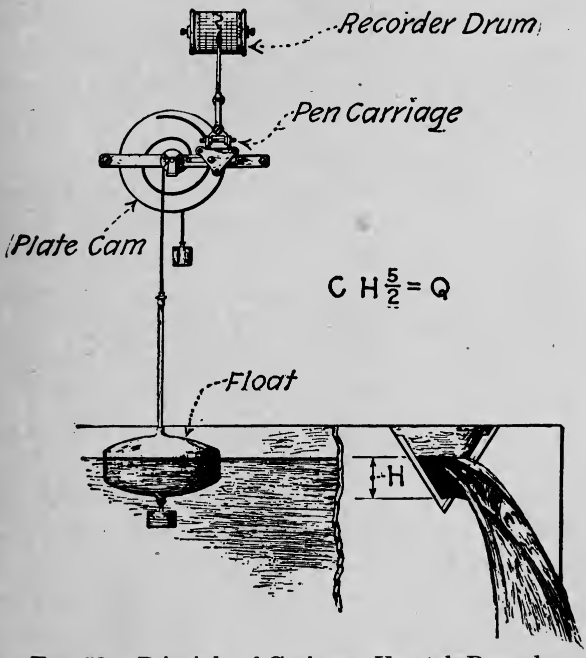


Fig. 58.—Principle of Cochrane V-notch Recorder.

is situated in a weir tank in such a way as to be in a quiet level. A vertical spindle on the float communicates the rise and fall of the water to a plate cam by means of a steel cable wound around a drum on the cam shaft. As the cam revolves, it engages a pin on a carriage so as to move the carriage from left to right according to the height of the float, and, therefore, the rate of flow. The

cam groove is a polar curve having the equation of flow; consequently the motion of the carriage is directly proportional to the flow and not to the head causing the flow. The carriage carries a pen which records on a drum chart.

The Lea V-notch recorder is much the same in general principle, but a cylindrical cam is used. Another type of V-notch recorder makes use of a specially shaped weighing float, the vertical motion of which is proportional to the rate of flow.

The equation cited under Test 19 for a 90° notch is $Q=2.53\sqrt{H^5}$ in which Q is in cubic feet per second, and H is in feet. If h is the head in inches, and q the cubic feet per minute, then

$$q=3.04\sqrt{h^5}.$$

These meters are generally graduated to read in pounds, in which case there may be an error because of change of density of the water. This error is partly compensated for, in that, at a higher temperature, for a given head, a lesser weight of water will pass, but, on the other hand, the float will stand lower in the hot water, thus making the recording pen indicate less. If the flow were proportional to the head, the correction would be exact.

(a) **Zero Level of Weir.** Some forms of these meters are provided with an inside and outside pointer indicating this level, the outside one being adjustable. If there is none, the hook gage can be set as described under Test 19, if convenient. When the hook gage, with which the calibration is to be made, is outside the weir tank (it is often located in a vertical pipe connected at the bottom with the tank, for convenience in handling, and securing still water), the following procedure may be used.

The weir is blanked off with some boards, as indicated by Fig. 59. A gage, g, of a convenient height, say 1 in., is set in the weir as indicated. The water level should now be brought to the point of this gage, and a reading of the hook gage taken. The zero level of the weir is then 1 in. below this reading.

(b) Calibration. Since the coefficient in the formula $q=3.04\sqrt{H^5}$ is very well established, and is practically constant under usual operation (ranging about 1 per cent above and below), this formula furnishes a ready means of calibration.

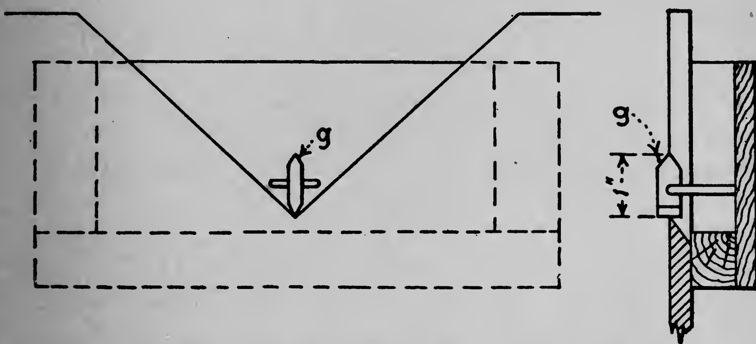


FIG. 59.—Gage Pointer for Finding Zero Level.

It is only necessary to measure the head corresponding to any instrument record, and from this calculate the actual flow. The mechanism should first be set to register zero when under a zero head as located by method outlined in (*a*). Note, that the height of the float should not be used for obtaining heads. Temperatures should be taken to enable the calculation of the flow in pounds.

The coefficient, 3.04, applies to a 90° notch. 54° notches also are used. This angle gives half the area of the right-angle notch. The coefficient is a little greater than half that of the latter, and may be taken equal to 1.55.

(c) Sensitiveness. With the hook gage set at a convenient height, start with a low level behind the weir, and gradually increase it until the gage is just submerged. At this instant a reading of the recorder should be noted. The procedure should be repeated, the hook gage position unchanged, with the head

decreasing. The difference between the readings represents twice the lag due to friction of the recording mechanism.

21. Calibration of a Nozzle

Principles. A nozzle makes a very convenient form of velocity water meter when the water to be measured may be discharged from a pipe into the atmosphere. The principle upon which this meter depends is similar to that of a weir, that for a definite pressure behind the opening there will be a definite flow through it. Then, if it is arranged that this pressure be measured, the flow may be ascertained in cubic feet or pounds per unit of time either by calculation or from a calibration curve.

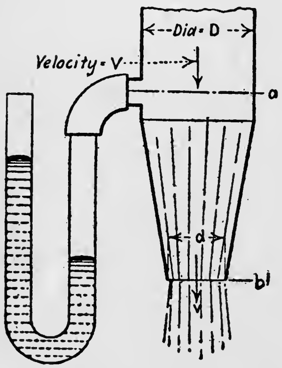


Fig. 60.—Nozzle.

The equation of a nozzle may be obtained from a consideration of the energy appearing in the water. In Fig. 60 the energy at the section a is in two forms, pressure and velocity, neglecting the small amount of potential energy due to the height at a above the opening at b. All the energy at a is converted into velocity at b, since the pressure at b is nil, except that used to overcome fluid friction between the points a and b. Let W equal the weight of water passing per second. Then the available energy at a is $WH+\frac{WV^2}{2g}$, H being the pressure in feet of water and V the velocity in feet per second at a. The expression $\frac{WV^2}{2g}$ is the kinetic energy at a, and WH the pressure energy. Likewise, the energy delivered at b is $\frac{Wv^2}{2g}$.

If v_1 is the velocity at b under the ideal condition of no losses, then

$$Cv_1 = v,$$

in which C is a coefficient less than unity. Since energy is proportional to the square of the velocity,

$$C^2 \times \text{available energy} = \text{delivered energy}.$$

Substituting in this the values of the energies as previously noted,

$$C^2\left(WH + \frac{WV^2}{2g}\right) = \frac{Wv^2}{2g}.$$

We also have the relation that the velocities are inversely proportional to the cross-sectional areas, or

$$\frac{V}{v} = \frac{\frac{\pi d^2}{4}}{\frac{\pi D^2}{4}} = \frac{d^2}{D^2}.$$

Combining these last two equations we have

$$v^2 = \frac{C^2 2gHD^4}{D^4 - C^2 d^4}.$$

Since the quantity, Q, in cubic feet per second, equals the velocity times the area of the stream,

$$Q = 0.7854 d^2 \sqrt{\frac{C^2 2gHD^4}{D^4 - C^2 d^4}}.$$

For convenience, this may be expressed

$$Q = 6.3C\,\frac{D^2}{\sqrt{R^4 - C^2}}\sqrt{H},$$

in which R is the ratio of D to d. Note that the units of D are feet.

The value of C for a well-designed nozzle is between 0.95 and 0.99. It is thus seen that such a nozzle may be used as a meter without material error if it is uncalibrated.

A convenient method of measuring the pressure is by a mercury manometer as shown in Fig. 60. The difference in level of the mercury in inches should then be multiplied by $\frac{13.6}{12}$ to convert into feet of water. The lower level of mercury should be referred to the datum plane through the end of the nozzle (when the nozzle is vertical) as this allows for the potential energy between the sections a and b which was neglected in the formula; the column of water in the right-hand leg of the manometer balancing that within the nozzle. If the mercury descends below the datum plane, the reading of the manometer must be corrected for the head of water between the lower mercury level and the datum. When the nozzle is horizontal, the datum plane should be through the axis of the nozzle.

(a) **Calibration at Various Rates.** The rate may be varied by turning a stop valve in the pipe to which the nozzle is affixed, and the rate may be measured by discharging the water into a calibrated tank or by weighing it. If the pressure measuring device is a manometer, the tube connecting it with the nozzle should be full of water as any air in it will cause the apparent water head to be different from the actual by amounts varying with the pressure. The water head below the datum plane may be corrected for by translating it from inches of water to

inches of mercury and subtracting from the manometer reading; or the manometer may be raised each time it is read so that the lower mercury level is at the datum plane at the instant of reading. The determinations of rate in the desired units should be plotted against pressures in inches of mercury.

Rules 4 and 5, page 8, should be carefully observed.

(b) The Coefficient of Discharge varies somewhat with the rate, at any value of which it may be found from the quantity formula, the values of H and Q and the diameters of the nozzle being known.

Problem 21_1. What is the discharge in cubic feet per second from a nozzle having a diameter ratio of 2 to 1, the smaller diameter being $\frac{1}{2}$ inch, if the manometer shows 5 ins. of mercury with its lower level 7 ins. below the nozzle mouth? Assume $C=0.97$. *Ans.*, 0.0246 cu. ft. per sec.

Problem 21_2. Assuming that a nozzle has been calibrated with water at 60°, will more or less water (weight and volume) emerge for a given reading of the U-tube when the water is at 120°? Why?

22. Calibration of a Venturi Meter for Water

Principles. The Venturi meter is similar in principle to the nozzle. It is, in fact, a nozzle discharging into a closed and properly shaped pipe instead of into the atmosphere. See **Fig. 61.** The water passing through the pipe shown carries a certain amount of energy in the form of pressure and velocity. When the water reaches the contracted section b, its velocity is increased, and therefore its pressure must be decreased, since the total energy, barring losses, remains constant. The drop in pressure between sections a and b thus becomes a measure of velocity and hence quantity.

To deduce the equation for the Venturi meter it is only necessary to equate the expressions for energy at the two sec-

tions a and b. Thus, if V and v are the velocities in feet per second, and H and h the pressures in feet of water at the sections a and b, respectively, and W the weight of water passing per second, then

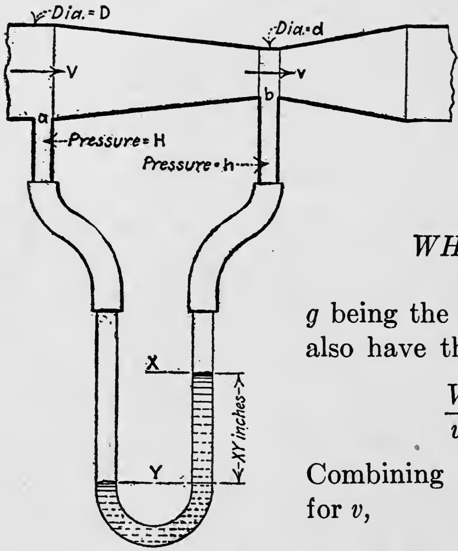


FIG. 61.—Venturi Meter.

$$WH+\frac{WV^2}{2g}=Wh+\frac{Wv^2}{2g},$$

g being the acceleration of gravity. We also have the relation

$$\frac{V}{v}=\frac{\text{Area at } b}{\text{Area at } a}=\frac{d^2}{D^2}.$$

Combining these equations and solving for v,

$$v=\frac{D^2}{\sqrt{D^4-d^4}}\sqrt{2g(H-h)},$$

from which the cubic feet per second discharged under ideal conditions is

$$Q'=\text{area}\times\text{velocity}$$

$$=0.7854d^2\frac{D^2}{\sqrt{D^4-d^4}}\sqrt{2g(H-h)}.$$

Owing to friction losses the discharge thus calculated is too high. To allow for this, a coefficient of discharge, C, less than unity, is introduced. Combining $\sqrt{2g}$ and 0.7854, for convenience, the formula becomes

$$Q=6.3C\frac{D^2}{\sqrt{R^4-1}}\sqrt{H-h},$$

in which R is the ratio of D to d, and D is in feet.

The value of C for a well designed Venturi meter is between

0.95 and 0.99, and the value of R is usually 2 or 3 to 1. It is thus seen that an uncalibrated Venturi meter may be used with not more than 2 per cent of error by assuming $C=0.97$.

The meter is generally shaped with an annular space A, Fig. 62, from which to lead the opening transmitting pressure. Between this annular space, or pressure chamber, and the interior of the tube are circular openings connecting the two. This arrangement transmits the pressure more truly than would a single opening.

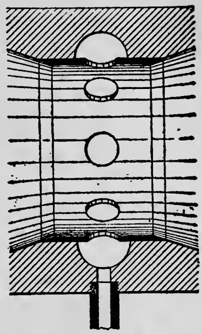


FIG. 62.

For measuring the difference of pressure, $H-h$, a mercury manometer is generally used connected as shown by Fig. 61. When the manometer is graduated in inches, the flow may be calculated or obtained from a calibration curve determined experimentally. The manometer is sometimes graduated in quantity rates so that no curve need be used.

Some forms of Venturi meter employ apparatus giving a continuous record on a time chart which, when integrated, yields total quantities. One of these is illustrated by Fig. 63. Note the cam by which movement of the mercury column (proportional to the square of the velocity), is rectified to give uniform radial chart ordinates.

(a) **The Calibration at Various Rates** may be made by controlling the rate by a stop valve in the water line and measuring the rate by discharging the water into a calibrated tank or by weighing it. A curve should be drawn between the rates and the differences of mercury level (manometer readings).

Rules 3, 4 and 5, page 8, should be carefully observed.

(b) **The Coefficient of Discharge** is determined from the formula at any rate when the values of Q, $H-h$, and the diameters of the instrument are known.

The head in inches of mercury should be transposed to feet of water. Referring to Fig. 61, it is seen that the column of water on the right above X balances an equal column on the left. The

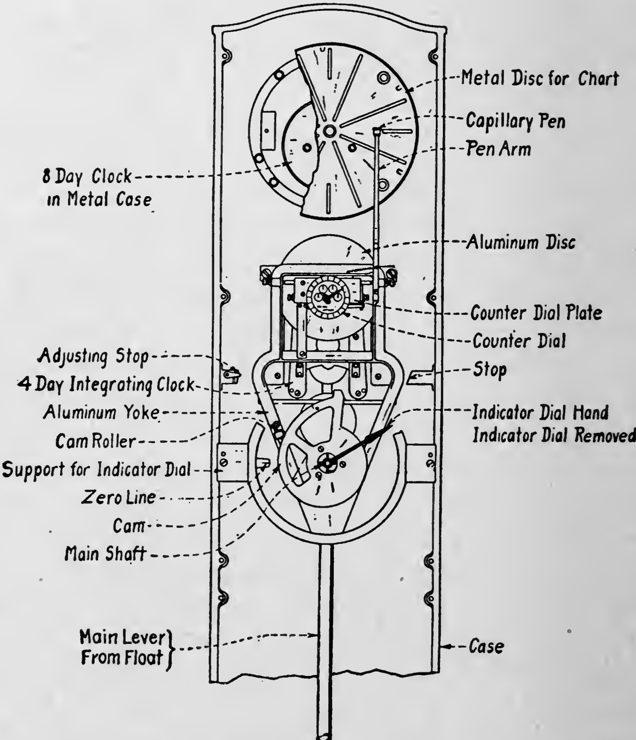


FIG. 63.—Builder's Iron Foundry Venturi Meter Recorder.

column of water between X and Y, however, is balanced by mercury, so that the pressure difference between sections a and b is not XY inches of mercury, but XY inches of mercury minus XY inches of water; that is

$$\frac{(13.6-1)}{12}XY = 1.05\ XY, \text{ feet of water.}$$

Problem 22_1. Given the diameter ratio 3 : 1; smaller diameter, 1 inch; find the constants K_1 and K_2 in the following equations:

$$Q = K_1 C\sqrt{H_1}$$

$$\text{Lbs. per sec.} = K_2 C\sqrt{H_1}$$

in which H_1 is the difference of level in the mercury manometer in inches.

Problem 22_2. Draw the calibration for the meter of Problem 22_1, assuming that $C=1$, up to a value of $H_1 = 6$ ins.

23. Calibration of a Venturi Meter for Gas

Principles. The deduction of the flow formula differs from that for water on account of the fact that gas carries intrinsic energy due to its expansive property which must be accounted for in the equation of energy. Otherwise the principle of the deduction is the same. The flow formula has been presented by Mr. E. P. Coleman in a paper to be found in the transactions of the A.S.M.E., Vol. 28, page 483. The general arrangement of the instrument is the same as for water. It has not been extensively used for gases probably because the Pitot tube (see Test 24) answers the same purpose more cheaply, is considerably smaller, and has been more developed experimentally. Nevertheless, there are obtainable and in successful use recording Venturi meters in large sizes for the measurement of gases and steam. One of these is similar to Fig. 63. To convert the records into standard cubic feet, corrections for both pressure and temperature must be applied.

(a) Calibration at Various Rates may be accomplished by using one of the methods described under Tests 25, 27, 28, or 29 for measuring the true quantity of gas; or if the Venturi meter is of small size, by using a gasometer. The rates may be obtained in cubic feet per second or minute, and plotted against difference of pressure in inches of water or mercury. Note that the temperature conditions must not materially vary from those in use, since a change of temperature is accompanied by a change of flow.

24. Calibration of a Pitot Meter for Water

Principles. Fig. 64 represents a Pitot meter, which is merely a curved tube placed in a stream of velocity V feet per second so that the immersed end of the tube faces the stream. The kinetic energy of a given weight of the water, W, is $WV^2/2g$, in which $V^2/2g$ is called the "velocity head" or the head of water, H, in feet, which under ideal conditions may produce a velocity, V. In the Pitot tube, Fig. 64, the water rises until the head in the tube just balances the velocity head of the stream. Hence

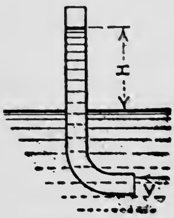


Fig. 64.
Pitot Tube.

$$V = \sqrt{2gH}.$$

The height H thus becomes a measure of velocity and therefore quantity.

If the stream is in a closed conduit, the water may be under a pressure greater than atmospheric which would cause it to rise in the Pitot tube to a height greater than that due to velocity. This may be allowed for by making a separate measurement of pressure. Thus, in Fig. 65, H_2 ft. of water balances velocity plus pressure, and H_1 ft. in the straight tube is due to pressure only owing to the fact that the velocity is not impressed upon this tube opening. The velocity is then

$$V = \sqrt{2g(H_2 - H_1)}.$$

The Pitot tube opening is called the "velocity opening" and the other, the "static opening." They are often connected to a differential gage as shown by the dotted lines of

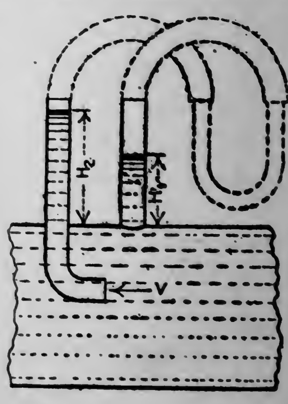


Fig. 65.—Pitot Meter.

Fig. 65 so that the pressure is balanced, and a single reading gives the velocity head direct.

The plane of the static opening should be parallel to the stream; for, if it is inclined toward it, the velocity is partially impressed; or, if away from it, suction will result. With a correct static opening there may be the same effects if the stream is not parallel to the pipe direction, which may be the case at points near bends or elbows, or when the instrument itself interferes with the regularity of the flow. Generally two static openings on diameters at right angles, connected together, are used.

When the Pitot tube and static opening are properly designed and applied, the actual condition of flow is the same as that represented by the formula, so that there need be no coefficient to determine as with the nozzle, weir, and Venturi meters.

In a closed conduit, the velocity of the stream varies through the cross-section, being maximum at the center and minimum at the sides. It is therefore necessary either to find the velocity at enough points to get a fair average or to read the instrument at a point of mean velocity. The mean velocity is approximately 0.83×velocity at center, and this rule is sometimes used for rough results.

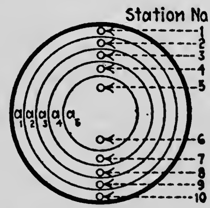


Fig. 66.

(a) Determination of Average Velocity and of Quantity Rates by Traversing. In Fig. 66, the concentric circles mark off equal areas so that $a_1=a_2=a_3=a_4=a_5$, the area of the pipe being $A=5a_1$. The average velocities in these areas are v_1, v_2, v_3, etc., respectively, the average velocity in the whole pipe section being V. Then, if Q represents the cubic feet per second,

$$Q=AV=a_1v_1+a_2v_2+a_3v_3+\text{etc.},$$
$$=5a_1V=a_1(v_1+v_2+v_3+\text{etc.}),$$

and

$$V=\frac{v_1+v_2+v_3+\text{etc.}}{5}.$$

That is, the average velocity in the pipe is the average of velocities taken at points representing equal areas.

The method to be pursued, then, is to take readings of the differential gage at such points for a single determination of quantity, the quantity being calculated by taking the product of the average velocity, V, and the area of the pipe. It is customary to take readings at ten points, or stations, as shown by Fig. 66. The readings on one side of the pipe center should duplicate those on the other, and there are two readings for each area. The stations should be located at the following distances from the pipe wall, D being the diameter of the pipe, in order to represent equal areas.

Station No. 1,	$.0256D$	No. 6,	$.658D$
No. 2,	$.0817D$	No. 7,	$.774D$
No. 3,	$.146D$	No. 8,	$.854D$
No. 4,	$.226D$	No. 9,	$.918D$
No. 5,	$.342D$	No. 10,	$.974D$

The calculation for V is simplified by taking the average of the square roots of the differential gage readings, and using this in the velocity formula for $\sqrt{H}$. Note that the head of the gage liquid should be converted into equivalent head of water in feet. If mercury is used as the gaging liquid, this may be done as described under Test 22 (b). If some other liquid, as oil, is used, its specific gravity should be taken into account according to the same principle. The quantity rate can be expressed as

$$Q = \text{a constant} \times \Sigma \sqrt{\text{gage reading}}$$

for the simplification of numerical work.

Recording Pitot meters in various forms are in considerable use, and of these a good example is that of the General Electric Co., applicable also to steam and gas flow (see page 128). These

instruments are calibrated with the velocity opening at a fixed position in the stream, and yield charted results in terms of pounds, gallons and cubic feet per unit of time.

(b) Location of the Point of Mean Velocity. If the pipe is traversed and the average of the square roots of the gage readings obtained as under (*a*), then the square of this average represents the gage reading at the point of mean velocity. This point may then be located by plotting on a chart gage readings against distances from the wall of the pipe; the distance corresponding to the square of the average square root of the gage readings being then taken from the chart.

The location may be found without plotting a curve by searching with the Pitot tube for the point at which is registered the gage reading as calculated above and then measuring the distance of the velocity opening from the pipe wall. If this is done, a center reading should be taken when making the traverse by which the uniformity of flow may be checked when the searching is done.

Problem 24_1. Deduce the values given in the table under (*a*) for the distances of the stations from the pipe wall.

Problem 24_2. Give a rough value of the quantity rate in cubic feet per minute in a 10-in. pipe (internal diameter = 10.02 in.) if the differential gage reading at the center is 26 ins. of oil the specific gravity of which is 0.9, an inverted U-tube being used. *Ans.*, 102 cu. ft. per min.

Problem 24_3. Figure the constant for use in the quantity formula for an 8-in. pipe (internal diameter = 7.98 ins.), the gaging liquid being mercury. *Ans.*, .285, in cu. ft. per sec. for 10 stations.

Problem 24_4. Figure the cubic feet per second in an 8-in. pipe (internal diameter = 7.98 ins.) if a traverse gives the readings, 1.02, 1.26, 1.42, 1.56, 1.69, 1.65, 1.5, 1.38, 1.25, 1.01 inches of mercury. *Ans.*, 3.32 cu. ft. per sec.

25. Constants of a Pitot Meter for Gas

Principles. The Pitot meter may be used for gas exactly the same as for water (see Test 24) except that greater precautions should be taken on account of the fact that gas is a much

more mobile fluid. The instrument should be placed at a distance of at least 12 pipe diameters from the nearest bend on the up-stream side, and four on the down-stream. A Pitot tube of unusual proportions or design should not be used without previous calibration, as it has been found by experiment that apparently unimportant details cause large errors in the indications. The best proportions of Pitot meter for gas under general conditions have not yet been satisfactorily established, but Fig. 67 represents the form recommended by the American Society of Mechanical Engineers. Numerous experiments, however, lead to the conclusion that it is sufficient to have two static openings at the wall of the pipe, instead of as shown by Fig. 67.*

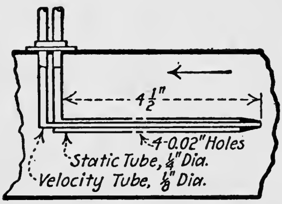


FIG. 67.—Pitot Tube for Air.

In the case of gas, in the formula

$$V = \sqrt{2gH},$$

H is the head due to velocity expressed in feet of whatever gas is flowing, allowing for its density due to its condition of pressure and temperature. The observed head in terms of inches of the gaging liquid must therefore be translated. For this purpose the following relation may be used,

$$H = \frac{h}{12} \times \frac{\text{density of gaging liquid}}{\text{density of gas in pipe}},$$

in which h is the observed head in the U-tube connected as shown in Fig. 65, expressed in inches. The densities are generally given in pounds per cubic foot. Water is usually used for the gaging liquid, the density of which may be taken as 62.3 lbs.,

* See work of Wm. Rawse, Vol. 35, Trans. A.S.M.E.

as it varies but little with temperature. Kerosene is also used. Its density may be figured from its specific gravity.

(a) **Determination of Average Velocity and of Quantity Rate by Traversing.** The experimental procedure is exactly the same as for Test 24 (*a*) when the gas is at room conditions of temperature and pressure, that is, approximately 70° F., and 14.7 lbs. Air, for example, under these conditions weighs .0749 lb. per cubic feet, so

$$H=\frac{h\times 62.3}{12\times .0749}=69.3h,$$

from which

$$V=\sqrt{2g(69.3h)}=66.7\sqrt{h},$$

in which h is the observed head in inches of water.

At other pressures and temperatures, the density varies according to the relation (for perfect gases)

$$144pv=p\frac{144}{w}=RT,$$

in which p = absolute pressure in pounds per square inch;
v = specific volume in cubic feet per pound;
w = density, in pounds per cubic foot;
R = 53.4, for air;
T = absolute temperature, degrees F.

From this relation, follows

$$w=\frac{144p}{RT}.$$

Now

$$V=\sqrt{2gH}=\sqrt{2g\frac{h}{12}\times\frac{62.3}{w}}.$$

Substituting the value of w and simplifying,

$$V = 11.1\sqrt{\frac{h}{p}\,T},$$

from which the quantity, Q, in cubic feet per second, may be figured by multiplying by the area of the conduit.

If the quantity rate, in pounds per second, W, is wished, the following may be used:

$$W = wQ = \frac{144p}{RT} \times .7854\,\frac{d^2}{144} \times 11.1\sqrt{\frac{h}{p}\,T},$$

in which d = diameter of conduit in inches. Simplifying,

$$W = .163d^2\sqrt{\frac{hp}{T}}.$$

The absolute temperature is figured by adding 460° to the temperature as obtained by a Fahrenheit thermometer. For the absolute pressure of the gas, p, the barometer should be read and added in the same units to the static pressure of the gas as shown by a U-tube or other pressure gage. This gage may be connected to the static opening of the Pitot meter.

For any other gas than air, or for any other gaging liquid than water, an equation may be deduced similar to the above by the same method.

The procedure, then, for gas at temperatures and pressures materially different from room conditions is either

First, to make a traverse from which the average of the square roots of the differential gage readings yields $\sqrt{h}$, and to get one set of readings of temperature, static pressure, and barometer.

Second, to place the Pitot tube at a point of average velocity, by which h is obtained by a single reading, other measurements the same.

(b) The Location of the Point of Mean Velocity may be found exactly the same as for water, Test 24 (*b*).

Problem 25_1. What is the velocity, cu. ft. per second, and pounds per second of flow of illuminating gas in a 3-in. pipe (internal diameter = 3.07 ins.) if the reading of the differential gage is 1.5 ins. of oil of specific gravity 0.85? Pressure of the gas is 4 ins. of water; barometer, 29.7 ins. mercury; temperature 55° F. *R*, for this gas, is 60.7. *Ans.*, 0.276 lb. per sec.

Problem 25_2. Figure the constant in a formula like $V=66.7\sqrt{h}$, to apply to Problem 25_1. Take average conditions to be 14.7 lbs. absolute pressure and 60° F. *Ans.*, 65.8.

Problem 25_3. What is the velocity in feet per second if $h=12$ ins. of water, pressure = 4 ins. of mercury, temperature = 120° F., barometer = 30.1 ins. of mercury, the gas being air. Figure answer by both approximate and accurate formulas. *Ans.*, 231 and 226 ft. per sec.

26. Calibration of an Orifice for Water

Principles. If water is caused to pass through an orifice, the pressure behind the orifice being measured, the flow may be calculated. Suppose that the velocity of the water behind the orifice is negligibly small; then the energy available is WH, W being in pounds per second, and H, the pressure in feet of water. This pressure energy is expended in imparting velocity, or kinetic energy, to the water upon passing through the orifice, so that, neglecting friction, etc., $WH=WV'^2/2g$, V' being the velocity in feet per second, and g the acceleration of gravity. Hence, under ideal conditions, $V'=\sqrt{2gH}$. Owing to friction losses, the actual velocity is somewhat less than this, so that if c_1 is a number less than one,

$$V=c_1\sqrt{2gH}.$$

If we multiply this by the cross-sectional area of the stream, we shall get the quantity, Q, in cubic feet per second. Now, this area is less than that of the orifice; the stream being contracted

by virtue of the tendency of the water particles to flow in the plane of the orifice; so that if c_2 is a number less than one,

$$\text{area of stream} = c_2 \times .7854\ d^2,$$

the orifice being circular and d ft. in diameter. It follows that

$$Q = c_1 c_2 \times .7854\ d^2 \sqrt{2gH}$$
$$= \quad c \times .7854\ d^2 \sqrt{2gH}.$$

c_1, c_2, and c are called the "coefficients of velocity, contraction, and discharge," respectively. Merriman gives average values for them as .98, .62, and .61 for the case of orifices with sharp edges, that is, orifices in thin plates beveled on the side not touched by the water, so that the water touches only one line of the orifice.

The coefficients vary with the head and with the diameter of the orifice.

In order that the velocity back of the orifice shall be negligible, the cross-section of the stream in the conduit should be at least one hundred times that of the orifice. If the water conduit is round, this means its diameter should be at least ten times that of the orifice.

The pressure back of the orifice may be measured by a mercury manometer or by a Bourdon gage, or the actual height of water above the orifice may be measured direct with a gage glass, float, or hook gage (see p. 96).

As with a rectangular weir, the coefficients vary considerably with both the head and the size of the opening. The average value, $c = .61$, should therefore be taken only for rough results.

The coefficient of contraction varies with the shape of the orifice, one having edges rounded toward the inside giving markedly less contraction.

(a) Determination of Quantity Rates at Various Heads may be made by discharging the water into a weighing tank,

or by measuring its volume. The quantity discharged in cubic feet per minute should be plotted against the head in the observed units.

(b) Determination of Coefficients. The value of c may be calculated from the formula when the values of Q, H, and d are known. The coefficient of contraction may be found by measuring the diameter of the stream at the contracted section with a caliper, this section being distant from the plane of the orifice by about a half diameter. Knowing this coefficient, and the coefficient of discharge, the coefficient of velocity is readily obtained.

Problem 26_1. What is the coefficient of discharge, if an orifice discharges 92 lbs. of water per minute under a pressure of 3 lbs. per square inch? Diameter of the orifice is 0.6 in. *Ans.*, 0.594.

Problem 26_2. If the diameter of the contracted section (see Problem 24_1) is 0.36 in., what are the coefficients of contraction and velocity? *Ans.*, 0.6; 0.99.

27. Calibration of an Orifice for Gas

Principles. The deduction of the flow formula differs from that for water on account of the fact that gas carries intrinsic energy due to its expansive property, which must be accounted for in the equation of energy. The theoretically correct flow formula is deduced in various works on thermodynamics. It includes necessarily a coefficient c to allow for contraction and losses, as is the case with water passing through an orifice. For gas, however, the value of c is very uncertain, especially for high pressures, there being insufficient experimental data on this quantity.

When the pressure drop between the two sides of the orifice is small, the volume of the gas changes but little, and consequently only little intrinsic energy is delivered to influence the flow. Under these circumstances, the intrinsic energy may be ignored, and an equation deduced similar to that for water (Test 62,

principles), namely $V=c\sqrt{2gH}$, the head, H, then being the height of a column of gas to produce the pressure drop, its density being taken into account.

Since the orifice generally may be designed of such size to produce a small pressure drop, it seems hardly worth while to use the accurate and much more complex formula, especially in view of the uncertainty of the values of the coefficient of discharge under those conditions to which the hydraulic formula does not apply.

The gas may be caused to flow into the atmosphere, in which case, the pressure drop equals the pressure of the gas above atmosphere, and may be read by a manometer. If the gas cannot be discharged in this way, the orifice may be in a plate fitting square across the gas conduit, and the drop of pressure measured with a differential gage as with the Venturi meter, Fig. 61, except that water should be used as a gaging fluid.

In any case, the cross-section of the entrance to the orifice should be of such size as to reduce the entrance velocity to a negligible value. If the orifice needs to be large to secure the desired pressure drop, and the conduit is relatively small, the purpose can be accomplished by making an enlarged section in the conduit.

In the case of the Pitot meter, Test 25 (*a*), velocity is measured by measuring a static pressure caused by it. In the case of an orifice, the procedure is reversed in that a static pressure, causing velocity, is measured and from it the velocity is determined. It follows that the derivation of the velocity formula is the same, and we have for an orifice after introducing the coefficient of discharge,

$$V=c66.7\sqrt{h}, \text{ for room conditions of air,}$$

$$V=c11.1\sqrt{\frac{h}{p}T}, \text{ for general conditions,}$$

and for the rate in pounds per second, d, being the diameter of the orifice in inches

$$W = c .163 d^2 \sqrt{\frac{hp}{T}};$$

the other notation being as given for the Pitot meter.

The value of c has been fairly well established for orifices in thin plates, up to about 5 ins. in diameter, and to a pressure of about 6 ins. of water. An average value of 0.6 may be taken within these limits with less than 2 per cent of error.

(a) Determination of Quantity Rates at Various Heads. The rate may be varied by varying the output or the speed of the machine handling the gas. If an air compressor or blower, the speed may be varied. If the orifice is used for such a purpose as measuring the exhaust from a gas engine, its external load may be varied, thus changing the amount of fuel and air used.

The true rate may be measured by any of the methods of Tests 25, 28, 29, or by a gasometer if the volume of gas is not too large.

Readings of the pressure difference between the two sides of the orifice should be plotted in the gage units against true rates.

(b) Determination of the Coefficient of Discharge. This is readily calculated from the flow formula, the calibration data of (*a*) being known. It is instructive to plot values of c against the pressure drop through each size of orifice, and against orifice diameters for each value of the pressure drop.

Problem 27_1. Write the equation of energy of air on the two sides of an orifice, neglecting kinetic energy at the entrance. Allowing a drop of pressure equal to 6 ins. of water, compare the velocity due to pressure energy with that due to intrinsic energy assuming adiabatic flow.

Problem 27_2. Design an orifice and containing conduit to measure the air supplied to a 12 H.P. gas engine taking about 20 cu. ft. of fuel gas per horse-power hour and about 12 cu. ft. of air to one of fuel. Drop of pressure through the orifice should not exceed 6 ins. of water.

28. Calibration of an Anemometer

Principles. An anemometer is a meter of the velocity type, generally consisting of a wheel with vanes against which the current of gas impacts causing a rotary motion proportional to the velocity of the current. This motion is transmitted to a gear and counter combination which registers linear feet continuously. By counting the time, the velocity in feet per second or minute is calculable.

When used to measure quantity, the anemometer is generally placed at the exit cross-section of the conduit discharging the gas. If the cross-section is rectangular it may be divided into a number of small squares defined, for convenience, by light strings or wire fastened from wall to wall of the conduit. The anemometer is placed in the middle of each of these squares and the velocity read. The average velocity through all of them may then be used with which to multiply the total area to get cubic feet per minute or second. If the conduit is round, the anemometer should be placed at points located as for a Pitot tube, Test 24 (*a*).

Anemometers generally are not adapted for velocities higher than 100 ft. per second, and are not very reliable. The vanes are apt to become deformed, causing false indications, and changes of frictional resistance of the bearings will have the same result.

When the velocities exceed the capacity of the anemometer, the discharge conduit may be enlarged in cross-section at the exit.

(a) Calibration against the Velocity of the Instrument in Still Air. The anemometer is mounted on one end of a horizontal arm 3 or 4 ft. long and pivoted at the other end so that the instrument may be caused to travel through the circumference of a circle. For this purpose the pivot may be supplied with a grooved wheel whereby the arm may be driven, through a belt,

by a motor or by hand. Knowing the revolutions per minute of the arm and its radius to the center of the anemometer, the linear velocity of the anemometer may be calculated. This is the true velocity of the air relative to the anemometer, and corresponds to the instrument reading. A number of such determinations are made at different velocities, and plotted as a calibration curve.

Care should be taken that the velocity is uniform throughout each trial, and that the error of starting and stopping is made sufficiently small. A small lever is generally arranged on anemometers by which the recording mechanism may be thrown in or out of gear. This may in some cases be operated while the instrument is moving.

(b) Curve of Correction Factors. Corrections may be figured as quantities in linear feet per minute to be added to or subtracted from the instrument indication for one minute. These should be plotted against linear velocities as shown by the instrument.

Problem 28_1. A blower discharges 2000 cu. ft. of air per minute through an 8-in. pipe. Design an exit conduit to reduce the velocity a proper amount so that the air may be measured with an anemometer, and show where the anemometer should be placed.

Problem 28_2. If the encircling frame of an anemometer is the same diameter as the opening through which gas is discharged, what should be the area with which to calculate quantity? Examine an anemometer to answer this question.

29. Testing a Calorimetric Apparatus for Measuring Gas

Principles. If gas flowing through a conduit is arranged to be heated or cooled by steam, water, or electric current, then, barring radiation from the conduit, the heat gained by the one medium equals that lost by the other. Thus, supposing the gas to be cooled by water pipes, if

W = weight of gas passing in a given time,
W_w = weight of water passing in same time,

T_1, T_2 = initial and final temperatures of the gas,
t_1, t_2 = initial and final temperatures of the water,
C_p = specific heat of the gas at constant pressure,

then

$$WC_p(T_1-T_2)=W_w(t_1-t_2),$$

from which,

$$W=\frac{W_w(t_1-t_2)}{C_p(T_1-T_2)}.$$

It is thus seen that with such an apparatus the air passing in a given time may be measured by weighing the water and taking the temperatures of the gas and water before and after cooling.

If steam is used in the coils, being condensed by the air, it is necessary to take the temperature of the water discharged and the pressure and quality of the entering steam from which its heat content may be obtained with the steam tables.

If electric current is used, the heat equivalent may be figured from voltmeter and ammeter readings.

The calorimetric method is sometimes useful for measuring large quantities of air or gas.

The Thomas Electric Gas Meter is an elaborate apparatus of this type, and is perhaps the most accurate device for measuring gas, especially in large quantities, on the market. Resistance thermometers are used, by which a constant temperature difference is maintained; and the current necessary to maintain this difference of temperature is measured in terms of standard cubic feet of air. The meter is entirely automatic and autographic.

An advantage of this type of meter is that no corrections for pressure and temperature are necessary, since the indications are proportional to weight and, therefore, to standard cubic feet.

(a) **Examination of Instruments.** The instruments should be sufficiently precise that the error of reading should be less than 2 per cent of the corresponding factor in the formula. The

weight of water may be measured readily with proper precision. The temperatures, however, appear as differences which may be only a few degrees. Hence, thermometers graduated to tenths may be necessary. It is sometimes useful to figure beforehand rough values of weights and temperature differences in order to ascertain the required precision of the instruments.

It is best, when using the apparatus, to take temperatures at various points in the cross-section of the gas conduit, to search for variations.

(b) Determination of Radiation Correction may be made approximately by stopping the flow of gas and maintaining the temperature within the conduit at an average value between the limits in actual operation. The heat necessary to maintain this temperature may then be figured for a unit of time. This may be used as a correction if very precise results are desired.

Problem 29_1. What should be the least count of an ammeter to measure 600 cu. ft. of air per minute under room conditions within 3 per cent of error? The voltage of the line is about 110. What should be the least count of the thermometers? Temp. rise of air to be about 20° F.

Ans., about 2 amp.

30. Calibration of a Steam Meter

Principles and Types. Steam meters are built on the Pitot, Venturi, and orifice principles. The quantity of steam flowing, in pounds per unit of time, is therefore proportional to the square root of a pressure or difference in pressure. The differential pressure, varying with the flow, actuates an indicating or recording device. When arranged to give a time chart showing rates and total quantities, the recording mechanism is generally rather delicate and complicated. It should be noted that with such meters, the steam flow is proportional not to the motion of the indicator or height of the chart, but to the square root of these quantities, unless some rectifying device is employed.

Changes in density of the steam, due to difference in pressure or superheat, should be allowed for, since the meters generally indicate the quantity in terms of pounds. This may be done by applying different tables, furnished by the makers, to interpret the indications, or by a hand adjustment of the recording mechanism. Variation of density due to wetness may be avoided by passing the stream through a separator just before it reaches the meter.

The General Electric Co. steam flow meter operates on the Pitot principle, the static and dynamic openings being made through a "nozzle plug" which is screwed into the steam main, different lengths being furnished for different diameters. The differential head is transferred through $\frac{1}{4}$-in. pipe filled with condensed steam, to a cast-iron, closed mercury manometer of the cup type, the rising column of which supports a float. On this float is a vertical spindle terminating in a rack. As the float rises and falls with the variation of pressure made by the steam flow, it revolves a magnet by means of the rack and a pinion. Another magnet, outside the mercury container (which, of course is under full steam pressure), follows the motion of the inside magnet; and this outside magnet motion actuates the indicator and recording pen mechanism. The graduations of the instrument are in arbitrary units from 1 to 10. To interpret the indications, it is necessary to multiply by $K_1 \times K_2 \times K_3 \times K_4$ determined, respectively, by the internal pipe diameter, quality and pressure of the steam, and the (exchangeable) size of the internal mechanism of the meter. Values of these quantities are furnished by the makers in the form of curves. Integration of the chart for total quantities may be accomplished with a special cam-operated planimeter which corrects for the variable ordinates.

The Curnon steam flow meter is a Pitot instrument, in which the differential pressure operates on a diaphragm set in a recorder case. The resulting motion of this diaphragm is communicated through an ingenious link-work to a pen-arm, rectifying the square

root motion so that the rise of the recorder pen is very nearly proportional to the steam flow. A feature of this instrument is that differences of density, accompanying variable steam pressure, are automatically corrected by a Bourdon tube.

A Venturi steam meter is made by the Builder's Iron Foundry and is, in principle, identical with the water meter illustrated on page 110. The General Electric Co. also apply the Venturi, instead of the Pitot principle, with the recorder previously described, for pipe sizes of 2 ins. and less.

The St. John steam meter uses the orifice principle at a constant difference of pressure and varies in orifice size to allow various amounts of steam to pass. This is accomplished by a float set in the orifice and so shaped that its motion varies the effective area of the orifice. The motion is transmitted to an indicating or recording device. In this case, the height of the diagram obtained is directly proportional to the steam flow.

A Simple Orifice Meter. When steam passes through an orifice, dropping in absolute pressure from P to p, the flow increases with the drop until $p=0.58P$, which is known as the "critical pressure." For lower pressures than this, there is no increase in the flow.

A convenient method of orifice measurement, when the steam is discharged at a pressure lower than $0.58P$, depends upon the application of Napier's formula, namely,

$$W=\frac{PA}{70}$$

in which W is the weight in pounds per second of dry saturated steam at an absolute pressure P lbs. per square inch, behind the orifice, and A is the area of the orifice in square inches.

Steam meters generally are not to be depended upon when the flow is very variable or intermittent. This condition may often be remedied by the location of the meter.

Integration of drum charts with uniform ordinates may be accomplished with the usual polar planimeter; but for circular charts the methods of Test 16 (*b*) may be applied.

(a) **Calibration** may be made by comparison with an accurate steam meter, or by direct weighing of the steam. The latter method is the more dependable. All of the steam passed through the meter in a given time should be condensed in a surface condenser and then weighed. A calibration curve may be made for each pressure or condition of superheat, if desired; the quantity of steam being varied by a valve in the steam line, between the meter and condenser. Generally it is sufficient to make the calibration at one condition of pressure, the steam being saturated. Throttling calorimeter readings should be made to insure the latter condition. The rate of flow should be kept as constant as possible during each of a number of runs at different rates, and the total condensate compared with the total obtained by integration of the chart, or average indications multiplied by time.

Before testing, the meter should be set correct at zero according to the maker's directions.

(b) **Sensitiveness of recording meters** may be examined by quickly stopping the flow of steam, and noting the time and character of the curve drawn when the pen returns to zero. Another useful test is to note the minimum amount of change in the valve opening regulating the flow, to produce a change in the recorder pen position. From this can be found the smallest change in the flow rate to which the meter will respond.

Problem 30_1. Using Napier's formula, what is the flow of steam in pounds per second through a $\frac{1}{4}$-in. diameter orifice, if the gage pressure is 30 lbs.? What is the lowest pressure to which this method applies if the steam discharges into the atmosphere? *Ans.*, 0.0314 lb. per sec.; 25 lb. gage.

Problem 30_2. Using a chart record from an actual steam meter, figure the average rate by taking off velocities at a number of equal time intervals. Figure the average rate from the mean height obtained by a planimeter. Compare results.

THERMOMETRY

Temperature may be defined as a measure of heat. It is purely relative and is generally dated from the freezing-point of water.

The expansion of materials, when heated, is approximately proportional to the heat added. It follows that the increase of length of a material is a measure of heat intensity. Upon this principle, the mercury thermometer was devised, mercury being chosen on account of its expansive properties. The Fahrenheit scale of degrees was chosen in a roundabout way by its inventor, who exposed his thermometer first to freezing water and then to water at the boiling-point. He marked the stem of the instrument at the level of the mercury in each case, and divided the distance between the marks into 180 equal parts. He continued the scale below the freezing-point by 32 of these divisions. The freezing-point thus became 32 degrees, and the boiling-point 212 degrees. Therefore the Fahrenheit degree may be defined roughly as the increase of temperature corresponding to 1–180 of the total linear expansion of the material chosen for a thermometer between the freezing- and boiling-points of water.

The material may be liquid, solid, or gaseous. Now, no known material expands uniformly in proportion to the heat added. The departure from uniformity, although slight, is appreciable in accurate work, especially at high temperatures. It follows that the divisions of a mercury thermometer, although absolutely equal, do not measure equal amounts of heat, and a degree of temperature is a quantity varying with its location on the scale. Also different materials expand according to different laws. Consequently, two thermometers of different materials, standing the same at the freezing- and boiling-points of water, will differ at all other points. It must not be thought

that this is because either one is inaccurate; it is simply that they are different standards. The linear expansion of a particular material, such as mercury, yields an entirely definite, though arbitrary, temperature scale, because the expansive properties of such a material, when pure, are constant.

In a mercurial thermometer, however, the degree is not measured by the expansion of mercury only; the glass containing it also expands. The scale, therefore, depends upon the relative expansion of mercury and glass. The latter varies considerably both in manufacture and expansive properties, so that for a standard scale it must be specified with great care.

Owing to the difficulty of obtaining glass of uniform quality, other materials than mercury have been preferred for accurate work. For the expansive material, air has been commonly used on account of its great expansiveness compared with that of the containing material. Nitrogen and hydrogen are now in more extensive use.

Gas thermometers are of two types: constant pressure and constant volume. The former uses the expansive properties of the fluid as in the mercury thermometer, while the latter measures the temperature indirectly by the pressure, which varies directly with the temperature according to Boyle's law. The resulting scales are different, but the differences are so slight that for most engineering purposes it is not necessary to discriminate between them. The gas scales are very nearly coincident with each other and with the theoretical "work scale" devised by Lord Kelvin. This latter is the ideal scale in which the degrees stand for equal amounts of heat. The simple mercury scale is lower than the others between the freezing- and boiling-points of water and higher above the boiling-point. The mercury-in-glass scale is higher than the others between the freezing- and boiling-points.

The nitrogen scale is the one commonly employed in scientific work, but the standard (between the freezing- and boiling-points)

is that of the constant-volume hydrogen thermometer. This was defined in 1887 by the International Bureau of Weights and Measures. The standard instrument is operated at an external pressure of one standard atmosphere, or 760 mm. of mercury, and the hydrogen is maintained at a pressure, when at the freezing-point, of one meter of mercury.

31. Calibration of High-reading Thermometers and Pyrometers

Principles and Types. Instruments for measuring temperatures above 600° F., are often referred to as pyrometers, but as there is no clear distinction between various devices denoted by this term and by the term "thermometer," the latter will be used in a generic sense. The writer prefers to use "pyrometer" for thermometers applicable to flame temperatures.

Mercury-in-glass thermometers are made for temperatures up to 1000° F. Their use, however, is limited because of their fragility and by the fact that the sensitive bulb and graduated scale are necessarily very close together.

Recording thermometers with disc charts have found much favor in power-plant work because of their convenience and accuracy. These are of the constant-volume type, working, in reality, as a pressure gage. The helical tube of the gage is connected by means of a flexible tube of small internal diameter to a bulb, and the system filled with a thermometric medium. Upon heating the bulb, the pressure of the medium is raised, transmitted to the helical tube, and recorded in terms of temperature. The working medium usually is a liquid for low temperatures, a vapor for medium, and a perfect gas for high. The first and the third employ charts with uniform scales, but the graduations of vapor (as alcohol) thermometers increase with the range. These are unaffected by changes of temperature of the flexible and Bourdon tubes. Such changes cause error with the thermometers

using liquids and gases, unless equipped with special compensators.

Another form of **expansion thermometer** takes advantage of the **relative expansion** of two metals. As they are heated, the motion of one relative to the other is magnified by a gear combination and transmitted so as to rotate a needle over a graduated dial, thus indicating the temperature.

The thermo-electric couple is one of the most accurate and convenient instruments for measuring temperature. It depends upon the fact that two wires of different metals having appropriate thermo-electrical properties will generate an electromotive force when connected at their ends, which electromotive force is proportional to the temperature difference between the ends. If it is arranged that the current be measured with a sensitive galvanometer, placed at any convenient distance from the thermo-electric "couple," we have an indirect measure of temperature. The galvanometer may be graduated in degrees or a calibration curve may be used to translate the electrical units.

In the case of thermo-couples, readings should be made of the temperature of the "cold junction," that is, the ends of the couple to which the galvanometer leads are joined. This is because the temperature indicated by the galvanometer is the difference between those of the cold and hot junctions. The cold junctions are often placed in ice to keep them at a constant known temperature. If a couple has been calibrated in this way against a standard couple, allowance should be made for the temperature of the cold junction when in ordinary use. This may be done by adding to the indications the difference between freezing temperature and the temperature of the cold junction.

For temperatures up to 1200 or 1400° F., the couple is composed of a wire of nickel and another of nickel and chromium. For temperatures up to 3000°, the metals are platinum and platinum with 10 per cent rhodium. The latter combination may be used for the low temperatures, but the former is preferred

on account of cheapness. Many other combinations have been used, but those cited above are the most usual.

The electrical resistance thermometer is one of the most accurate of temperature-measuring devices. This consists of a length of wire material, capable of resisting the heat, as platinum or nickel, the resistance of which can be measured with a Wheatstone bridge. When the temperature of the wire increases, its resistance goes up, so that the one can be measured by the other, when the coefficient of resistance is known (that is, the change of resistance per degree of temperature). This coefficient may be determined with great accuracy, and from it can be plotted a calibration curve of temperatures against resistance.

Optical pyrometers are advantageous when the hot body to be measured is inaccessible to a couple or bulb. One of them, the Morse, depends upon a comparison of the brightness of the hot body with that of an electric filament which is caused to glow more or less brightly by varying the current passing through it. Then, by measuring the current, an indirect measure of the temperature is obtained as with the thermo-electric couple. When used to measure furnace temperatures in boiler testing, a thin plate of iron is hung at the required point and its brightness is compared with that of the filament. This type of pyrometer is not very precise because it is difficult to judge exact similarity of brightness.

Another optical pyrometer depends upon the thermo-electric couple principle, but the couple is acted upon by radiant heat concentrated upon it by means of lenses in the form of a telescope.

There are other forms of pyrometers using other principles, but the ones listed above are the principal ones.

There are three ways generally used to calibrate high-reading thermometers and pyrometers: first, by comparison with a standard or secondary standard instrument; second, by comparison with the temperature of saturated or wet steam determined from

its pressure; third, by comparison with temperatures as shown by the melting-points of metals or boiling-points of fluids and salts.

(a) **Calibration against a Standard.** For this purpose, mercury thermometers may be used for temperatures up to 1000° F. Above, and often below that temperature, a favorite instrument is the thermo-electric couple, as it is one of the most accurate and easy to use.

The temperature is varied in a gas or electric furnace in which the bulbs of the standard instrument and the one to be tested are placed. Care should be taken to bring the whole furnace up to a uniform temperature before recording observations, by allowing sufficient time for heating. The sensitive portions of the two instruments should be placed as close to each other as possible to avoid errors from localized temperatures. It is best to take a series of decreasing readings to supplement the increasing ones, allowing the furnace to cool for this purpose. If the up and down readings are different at a given temperature, as they will be with certain types of instruments, they should be averaged. (See Test 2, principles.)

(b) **Calibration against Steam Temperatures.** This method is limited by the available pressure. At 200 lbs. gage, the temperature is 388° F., and at 500 lbs., about 470 degrees. It is thus apparent that not very high temperatures can be reached with saturated steam.

The procedure is to vary the temperature in a steam chamber similar to that used for testing indicator springs, Fig. 33, page 54, by controlling the pressure. The thermometer is inserted in a well set in the chamber and readings taken of it and of an accurate and sufficiently precise pressure gage at each variation of the pressure. The actual temperatures are ascertained by reference to the steam tables. A barometer reading is necessary to get absolute pressures.

When the steam is throttled to reduce its pressure it may become superheated, in which case its temperature is not deter-

minable. To avoid this, water may be sprayed into the steam before it enters the steam chamber, or the chamber may be water-jacketed. It is essential that there should be definite knowledge of the steam's wetness, for which purpose a throttling calorimeter (see Test 32) may be referred to.

A calibration of an instrument indicating beyond the available temperatures may be had by exterpolation. That is, after the experimental data have been plotted, the curve is continued beyond the known points. If the curve is a straight line, the result may be good, but caution should be used not to extend the exterpolation too far.

(c) Calibration against Melting- and Boiling-points. This method will be found quite convenient for ranges between 200° and 800° F. Its accuracy depends upon the purity of the materials.

When using the liquids and salts, they may be boiled in Florence flasks over a Bunsen burner, and the thermometer should preferably touch the liquid as well as be surrounded by the vapor. The following materials may be used:

Water, boiling at	212° F.
Aniline, boiling at	365° F.
Naphthaline, boiling at	420° F.
Diphenylamine, boiling at	590° F.
Anthracine, boiling at	664° F.
Zinc, melting at	786° F.
Sulphur, boiling at	832° F.
Antimony, melting at	1167° F.

(d) Stem Corrections. If a mercury thermometer is to indicate correctly, all of the mercury, both in the stem and bulb, should be subjected to the measured temperature. In practice this is rarely the case, as part of the stem containing mercury is at or near room temperature. For accurate work it is then necessary to apply a "stem correction."

Let S = stem correction in degrees;

H = height of column above actual position of mercury, if uniformly heated, inches;

l = length of one degree on the stem, inches;

N = number of degrees on the stem exposed to room temperature;

T = temperature, Fahrenheit, of the bulb;

t = average temperature, Fahrenheit, of the exposed mercury and stem;

C = cubical coefficient of expansion of the glass and mercury, combined by subtracting the value of one from that of the other.

Then

$$H = C(l \times N)(T - t).$$

Also

$$S = \frac{H}{l}.$$

Hence

$$S = CN(T - t).$$

The value of C for the Fahrenheit scale may be taken as 0.000088 and for the Centigrade scale, 0.00016.

It is generally sufficiently accurate to take the actual reading as the value of T. If desired, the corrected temperature thus obtained may then be used as T, and a second and more nearly exact stem correction calculated.

The value of t is somewhere between room temperature and that of the bulb. It is often estimated by hanging a second thermometer against the one to be corrected, with its bulb at a middle point on the exposed mercury column. It is very doubtful that this yields a close value for t.

Whether or not a stem correction is necessary depends upon the purpose for which the temperature is measured. In engineering work, thermometers are generally used to measure temperature differences, so that the percentage of error in the result

may be based upon a much smaller quantity than the actual number of degrees read. In many cases, when the temperatures are comparatively low or when the stem is well immersed, the correction is negligible.

Problem 31_1. A thermometer reads 240° when it is immersed to the 60° graduation. If temperature of room is 80°, what is the actual measured temperature? Estimate the stem temperature as an average of room and indicated temperatures. *Ans.*, 241.4°.

Problem 31_2. What should be the least count of a pressure gage to calibrate by steam a thermometer whose least count is 2 degrees? Why? *Ans.*, 1 lb. at 300°.

Problem 31_3. Account for the differences between up and down readings in the calibration of a thermo-electric couple.

HEAT OF STEAM—CALORIMETRY—SAMPLING

The English measure of heat is the British thermal unit, or B.t.u. This is loosely defined as the amount of heat necessary to raise the temperature of 1 lb. of water 1° F. As the specific heat of water varies with the temperature, a precise definition should name the temperature range. Several ranges have been used, as 32 to 33 degrees, 39 to 40 degrees, 61 to 62 degrees, so that there are a number of values for the B.t.u. The present tendency favors the "mean B.t.u." which is the average amount of heat per degree required to raise a standard pound of water from the freezing- to the boiling-point. One advantage of this unit is that it is independent of the temperature standard (whether gas or mercury) since it is fixed by the two temperatures which are the same on all thermometric scales. This is the unit used in the Marks and Davis' steam tables, values from which will be used in the examples in this work.

The heat of steam is counted from a temperature of 32° F. and is the amount of heat necessary to convert 1 lb. of water

at that temperature into steam under the given physical conditions. This includes the heat added to the liquid to bring it to the boiling-point at the given pressure, and the heat then necessary to vaporize it. If the steam is superheated, still another amount of heat is added to it to raise its temperature from that at the boiling-point, the pressure remaining the same. All of these heat quantities depend upon the pressure of the steam. It is thus seen that the physical conditions of steam determining its heat content are, first, its pressure, and second, its wetness or superheat. For example, the heat of the liquid, referred to as h, of steam at 14.7 lbs. absolute, is 180 B.t.u., being the heat necessary to raise the temperature of 1 lb. of water from 32 to 212°; 970.4 B.t.u. are then necessary to vaporize it. The heat of vaporization is referred to as L. Thus the total heat, H, necessary to make 1 lb. of dry steam at 14.7 lbs. is $h+L=1150.4$ B.t.u. If less than the whole pound of water has been evaporated so that there is $1-x$ lb. of water as a mixture, x being the part of a pound actually evaporated, then

$$H=h+xL.$$

This is the generic expression for the heat added to make 1 lb. of wet steam in which x may have any value up to unity.

For superheated steam the generic expression is

$$H=h+L+C_p(T-t_s),$$

in which C_p is the specific heat of superheated steam, t_s is the temperature of the dry or "saturated" steam at the given pressure, and T is the temperature of superheat. The value of C_p varies with T and with the pressure.

These various physical conditions are represented by Fig. 68, which shows 1 lb. of H_2O in a cylinder with a movable end under a pressure P.

It should be remembered that the datum temperature of 32° is taken purely as a matter of convenience, and that the actual amount of heat contained by steam, or for that matter any material, is a relative quantity. The significance of heat quantities so counted enters when we take the difference between two of them representing two conditions of H_2O. Thus the heat per pound given up by a boiler in making steam is the difference between the heat content of the H_2O leaving the boiler, counting from 32°, and that of the feed-water entering, counted similarly.

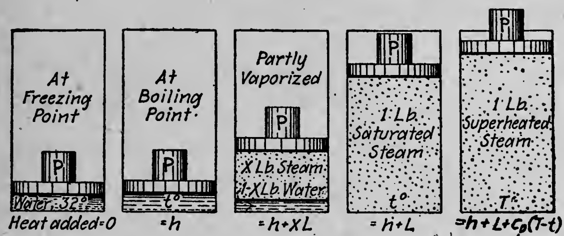


Fig. 68.—Heat Added to One Pound of H_2O.

To measure the heat of a given quantity of steam, it is necessary to know the amount in pounds and the heat content, H, or heat added per pound; the one multiplied by the other gives the heat of the total quantity. The weight of steam may be obtained by a steam meter, Test 30, or by any of the methods given under engine or boiler testing. The steam tables are used to get the value of H, since they list the heats of liquid and vaporization at different pressures. If the steam is dry, it is sufficient to take its pressure or temperature for reference in the tables. If it is superheated, its pressure should be read, and its temperature, T. The steam tables furnish values of t_s, and C_p, from which H may be calculated, knowing also h and L. The steam tables also contain heat contents of superheated steam, so that the calculation is not always necessary.

If the steam is wet, the value of x, as well as the pressure or temperature, must be known to fix its heat content. This is the condition generally met in practice, which makes determinations of wetness or "quality" important, especially for engine and boiler tests. (See Steam Tables pp. 371–373.)

The Heat of Steam when Mixed with Perfect Gases. If the pressure of the steam were known, its total heat could be found by reference to the steam tables as previously described. When steam is mixed with other gases, however, its pressure is less than that of the mixture. This follows from Dalton's law, which may be stated thus:

In a gas mixture occupying a given volume, the pressure of each constituent gas is the same as though it occupied the volume alone, and the pressure of the mixture equals the sum of the pressures of the constituent gases.

For example,

1 cu. ft. of air at 60° F., and 14.7 lbs. pressure weighs 0.0764 lb.
1 cu. ft. of steam at 60° F., and 0.256 lb. pr. weighs 0.00083 lb.
1 cu. ft. of mixture has a pressure of 14.956 lbs., weighs 0.07723 lb.

The figures here given for the steam apply to the saturated condition. At the temperature named, no greater weight of steam than 0.00083 lb. could exist in one cubic foot of volume, whether that volume were also occupied with another gas or not, because if there were more than this amount of H_2O present, its density would be greater than that of saturated steam at 60°, and consequently some of the H_2O would be in the form of water. On the other hand, at this temperature, there may be less than the named amount of steam present. In such a case, the pressure of the steam is less than 0.256 lb., since there is less of it, *and therefore it is at a temperature higher than that due to saturation; that is, the steam is then superheated.* This is the form in which humidity in the atmos-

phere usually exists. It is also the form in which H_2O appears in the products of combustion of fuels.

Consider now how the total heat of superheated steam in such a mixture may be determined. Let P and T stand for the absolute pressure in pounds per square foot and the absolute temperature in degrees Fahrenheit respectively, of the mixture. These quantities are readily measurable. Also it is possible to figure the weight of one cubic foot of the gases exclusive of the steam, and the weight of steam in one cubic foot of the mixture. Now the pressure of the gases exclusive of steam may be figured from the relation $P_1V_1=RT$, in which P_1 is the desired pressure, V_1 is the reciprocal of the weight of the gases per cubic foot, and R may be obtained from works on thermodynamics. (For air, $R=53.4$.) Then the pressure of the steam equals

$$P-P_1.$$

Knowing the pressure of the steam, values of h, L, and saturation temperature, t_s, may be found from the steam tables; which values, together with the temperature of superheat, T, make possible the calculation for total heat, H, according to the expression previously given. (See also "Hygrometry," Appendix.)

This calculation is laborious, so the following method, sufficiently accurate for engineering purposes, is preferable. The total heat of steam in the form of humidity in air or gases of combustion equals $1058.7+0.455\ T_e$, in which T_e is the temperature in degrees, Fahrenheit, of the mixture. This is an empirical relation proposed by Professor Diederichs.

In the analyses of boiler, gas engine, and gas producer trials, the heat of the steam in the exhaust gases, counting from room temperature, is required. The heat in the H_2O before vaporization is $t-32$, t being the room temperature. Subtracting this from the expression just given, we have, roughly,

$$1090+0.46\ T_e-t.$$

Steam calorimeters are instruments for measuring the quality of wet steam. Various forms are made depending upon different principles. These depend either upon a transfer of heat by which it can be equated to a measurable quantity or upon the mechanical separation of the entrained moisture. The first is strictly a calorimetric process. Thus if a sample of the steam whose wetness is to be measured is condensed in water, the heat given up to the water is readily measured, and this quantity can be equated to $h+xL$, in which x is unknown. The equation then yields the value of x. Another method is to superheat the steam, in which form its heat content is readily measured.

The mechanical separation method uses an apparatus similar to the ordinary steam separator, and involves weighing both the steam and the separated water.

Sampling. As usually arranged only a small sample of the total steam is tested for a quality determination. The accuracy of the result is primarily dependent upon the representativeness of the sample, so that every care should be used to secure a correct one. Unfortunately, it is always uncertain that a reasonably accurate sample has been obtained, but by using the proper shape of sampling pipe, inaccuracy may be reduced to a minimum.

In a horizontal steam pipe, water is apt to run along at the bottom, separated from the steam. If all of this water is passed into the sampling pipe, an undue amount of water appears in the sample. In a vertical pipe moisture runs down the inside of the pipe wall. Undoubtedly some of this should be included in the sample, but it is no easy matter, theoretically or practically, to secure the right proportions. Fig. 69 shows an approved form of sampling tube.

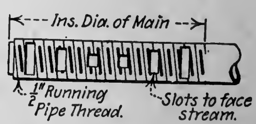


FIG. 69.—Sampling Pipe.

32. The Throttling Calorimeter

Principles. Fig. 70 represents the instrument diagrammatically. Steam from a sampling tube enters the steam pipe from which it passes through the orifice O, about $\frac{1}{16}$ in. in diameter, into the chamber C, which is open to the atmosphere. The steam is throttled upon passing through the orifice, and drops to a pressure only a little in excess of atmospheric. Now the heat contained by 1 lb. of dry steam at high pressure is greater than that at low pressure. Upon reaching the chamber, C, the steam therefore may contain an amount of heat in excess of that necessary for saturation, and this excess goes to evaporate the moisture carried in with the steam and to superheat both. In this condition, the heat content is readily measured, which makes possible a heat equation involving one unknown. Before entering the orifice, the heat of 1 lb. of the H_2O is h_1+xL_1. (See page 140 for notation.) In the calorimeter chamber, it is $h_2+L_2+C_p(T_2-t_2)$. Neglecting radiation,

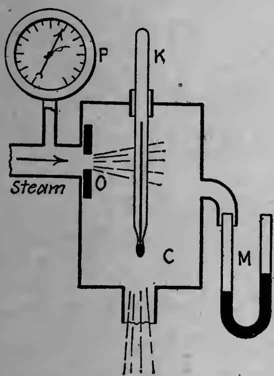


Fig. 70.
Throttling Calorimeter.

$$h_1+xL_1=h_2+L_2+C_p(T_2-t_2).$$

from which, $$x=\frac{h_2+L_2+C_p(T_2-t_2)-h_1}{L_1}.$$

The values of h and L and t_2 are found from the steam tables, by reference to the pressures indicated by the pressure gage P and the manometer M, Fig. 70. A barometer should be read to reduce these pressures to absolute. The temperature T_2 is obtained from the thermometer K. At atmospheric pressure C_p equals 0.47, but for rough and ready calculations it may be taken as 0.50.

Certain approximations in the use of the above formula may be made to simplify calculations, which, in view of the uncertainty of sampling, give an ample degree of accuracy. Thus, the pressure in the calorimeter chamber generally is only a few inches of mercury, above atmosphere. If it is taken as 14.7 lbs. absolute, $h_2+L_2=1150$, and $t_2=212$. This obviates the use of the manometer and barometer. Then the equation may be written

$$x=\frac{1150-h_1}{L_1}+\frac{.47(T_2-212)}{L_1}.$$

If the pressure on the entering side fluctuates but little, say 5 or 10 lbs., constant values of h_1 and L_1 may be applied. Now, a value of x may be calculated for $T_2-212=0$, taking h_1 and L_1 at an average pressure. The increment of x corresponding to one degree of superheat should then be figured. It is then a simple matter to add to x (when $T_2-212=0$) the increment corresponding to any number of degrees of superheat.

For example, when the average pressure is 100 lbs. gage $h_1=308$ and $L_1=880$. With no superheat, $x=(1150-h_1)\div L_1=(1150-308)\div 880=0.956$. To this must be added for each degree of superheat an amount equal to $0.47\times(1°)\div 880=0.00053$. If the thermometer in the calorimeter chamber shows 222°, the superheat is 10° and the quality of the steam is $0.956+10\times 0.00053=0.961$, and so on. This method of calculation is convenient for use during an engine or boiler test, since the quality is readily figured from the thermometer reading only, once the constants as obtained above are known.

The errors of the throttling calorimeter are due mainly to false thermometer readings of the superheat, and to radiation of heat from the instrument or fittings. It is best to set the thermometer in the calorimeter without using a well, which may be readily done with a perforated plug of wood or rubber. The thermometer should be moved vertically in the calorimeter

until the hottest part is found when the steam is passing as in regular use. To avoid radiation, all the parts should be well lagged. Not too large an orifice should be used, as this raises the back pressure in the calorimeter, and passes more steam than necessary.

(a) **Comparison of Indications** of various types of throttling calorimeter. Set up a number of calorimeters of different construction and heat protection so that they will all receive steam of the same quality. Then compare the temperatures of superheat, and the qualities of steam as shown by each

Fig. 71 shows a "jacketed" calorimeter designed to avoid error from radiation of heat. The main orifice is at A. Steam also enters the annular space D through another orifice B, thus keeping the walls of the calorimeter chamber C at the same temperature as the steam inside. An advantage of this type is that there is no back pressure on the calorimeter chamber and therefore no manometer need be used.

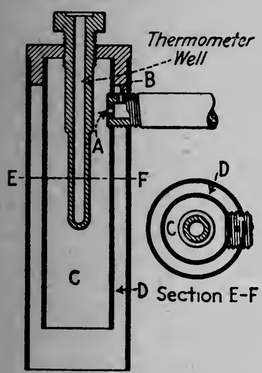


FIG. 71.

This instrument may be used as a standard and the errors of other calorimeters, lagged or unlagged, determined by comparison of their indications with those of the jacketed calorimeter.

The equation of the throttling calorimeter is made on the assumption that the heat content of the steam entering is the same as that in the calorimeter chamber. The effect of radiation of heat is to make this assumption untrue, and the thermometer indicating the temperature of the steam in the calorimeter chamber will read lower than if there were no heat radiation from the calorimeter chamber.

It is particularly important to note that when the thermometer in the calorimeter chamber indicates 212 degrees or less, the quality of the entering steam is indeterminate.

In appendix No. 11 of the A. S. M. E. Power Test Code, paragraphs 261 and 262, is given a method of correcting for radiation in the use of a given calorimeter by taking a "normal" reading of the thermometer when the instrument is arranged to receive actually dry steam. The difference between the "normal" reading and the calculated temperature of superheated steam, assuming the entering steam saturated, is the correction to be applied in use. This method is open to criticism on two scores: there may not be complete dryness when the normal reading is observed; and the conditions during use may be different from those prevailing during the determination of the correction.

Problem 32_1. Pressure of steam in the calorimeter chamber is 1.5 ins. of mercury. Barometer is 30.3 ins. of mercury. If the steam were saturated in the chamber, what would the temperature be? If the thermometer indicates 280°, how many degrees of superheat are there? What is the total heat of the steam in the calorimeter at 280°? If the pressure of the incoming steam is 84 lbs. gage, what is its quality? *Ans.*, $x = .995$.

Problem 32_2. Figure the quality from the data given in Problem 32_1, assuming the calorimeter pressure to be equal to atmospheric, 14.7 lbs. What is the percentage of error?

Problem 32_3. Figure the quality of steam corresponding to superheats of 25, 45, and 65°, using the approximate method of constants for Problem 32_1. *Ans.*, .972, etc.

Problem 32_4. What is the maximum degree of superheat that it is possible for the steam in the calorimeter to attain if the steam pressure is 95 lbs. gage, and the calorimeter pressure 14.7 lbs. abs.? What is the maximum if the steam pressure is 65 lbs. gage?

Problem 32_5. What is the maximum percentage of moisture that can be shown by a calorimeter under the two conditions named in Problem 32_4? If the calorimeter pressure is reduced to 10 lbs. abs. by connection with a condenser, what then is the maximum percentage of moisture?

Problem 32_6. If a thermometer reads 3° low on account of radiation, what is the percentage of error resulting from ignoring the radiation correction? Use the data of Problem 32_1. *Ans.*, 0.17%.

33. The Separating Calorimeter

Principles. If the water entrained in steam is mechanically separated, and the weights of the dry steam and separated water are then measured the percentage of water may be readily calculated. Mechanical separation may be effected by deflecting

the steam and water through a sharp bend; the water is then thrown out by centrifugal force.

Fig. 72 shows diagrammatically the Carpenter separating calorimeter. The water collects in the calibrated chamber C and its amount is measured by the scale S placed against a gage glass. The weight of dry steam is measured by the orifice method according to Napier's rule (see p. 116), the orifice being located at O. Since the weight of steam discharged from an orifice into the atmosphere equals a constant times the absolute pressure, an ordinary pressure gage G may be calibrated to indicate the weight of steam passing per minute. In operation, a steam sample is passed through the calorimeter, and, after it is thoroughly heated, initial and final readings of the water level are made for a measured interval of time, together with a number of readings of the gage from which an average is calculated. The gage gives the pounds of dry steam passing per minute; multiplying this by the number of minutes gives the total weight of steam. This weight is then divided by the sum of the weights of water and steam to get the quality.

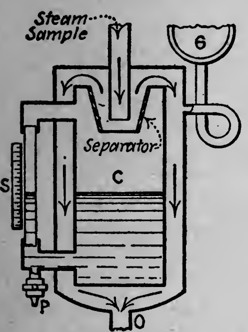


FIG. 72.—Separating Calorimeter.

The chief advantage of the separating over the throttling calorimeter is that it is operative no matter how low the quality. Experiments have shown that complete separation of the water may be depended upon with a properly designed separator.

(a) Calibration of the Dry Steam Meter may be made as for Test 30 (*a*). A calibration curve for the gage should be plotted.

(b) Calibration of the Water Gage. The chamber C, Fig. 72, is allowed to fill with water separated from steam passing as in usual operation. A small amount of this water is then drawn off from the pet cock P into a flask of cold water (to prevent

evaporation). The weights of the flask and contents and the heights of the levels in the water gage before and after the operation are carefully measured. A number of such determinations furnish the data for a calibration curve.

If the water used for the calibration is cold, its weight should be corrected for the difference in density due to the different temperatures. The temperature in usual operation may be taken as that of the steam passing through the calorimeter. (See Appendix for density of water.)

(c) Determination of Radiation Correction. Radiation from the instrument may cause a greater amount of water to be thrown down than was in the sample. In the Carpenter instrument, on account of the steam jacket arrangement (see Fig. 72), radiation occurs after the steam has left the separator; the water gage then remains correct, but the orifice measurement may be faulty since the orifice passes wet instead of dry steam. If the gage has been calibrated for the particular conditions of radiation, this calibration takes care of the radiation correction.

If radiation causes an increase of water in the water gage the correction may be determined as follows: Dry steam should be supplied to the calorimeter either with an apparatus as described under Test 32 (*a*) or by taking the sample from another steam separator. Any water that shows at the water gage of the calorimeter is then due to radiation. If it is figured as so many pounds per minute, the correction in ordinary use is readily applied.

Problem 33_1. If the water gage has been calibrated using the water separated from steam at 100 lbs. gage, what percentage of error will there be when the instrument is used for steam at 50 lbs. gage, the error being due to the change in density of the water between the two temperatures? If the water in the gage glass is 150° F., due to radiation, and the water in the calorimeter chamber is 325° F., will the level in the glass be above or below the inside level? Will these errors make material error in the results of x and why? *Ans.*, 2.4%.

Problem 33_2. If the orifice of the separating calorimeter is used as part of a throttling calorimeter so as to allow for moisture escaping the separator,

deduce a formula for the quality of the steam in terms of the weights shown by the separating calorimeter and the quality (x_1) as shown by the throttling calorimeter.

34. The Condensing Calorimeter

Principles. If a sample of steam the quality of which is to be determined, is condensed either by mixing with cool water, or by entering condensing coils surrounded by water, the arrangement constitutes a condensing calorimeter.

Let W_s = weight of condensate, including moisture in sample;
h, L, x = heat of the liquid, latent heat, and quality of the steam sample;
W_w = weight of condensing water;
t_w = temperature of water before it is heated;
T_w = temperature of water after it is heated;
T_s = temperature of the condensate.

If condensation is accomplished by mixing, $T_w = T_s$, otherwise they have different values. Now, if the calorimeter operates continuously, and if radiation is neglected

$$W_s\left\{(h+xL)-(T_s-32)\right\} = W_w(T_w-t_w).$$

That is, the heat lost by the steam equals that gained by the water.

If the operation is not continuous, the material of the calorimeter absorbs an amount of heat which must be accounted for. This quantity is called the "water equivalent," being the weight of water that absorbs the same amount of heat for a given temperature rise as does the calorimeter. For non-continuous condensing calorimeters, the water equivalent should be added to W_w.

The equation may be solved for x, the other quantities being obtained experimentally. h and L are found from the steam tables, the pressure of the steam being observed.

A barrel of water on a platform scales makes a non-continuous mixing calorimeter. An injector may be used as a continuous mixing calorimeter. A surface condenser may be arranged as a continuous non-mixing calorimeter.

(a) **Determination of the Water Equivalent.** The condenser should be isolated and filled with water either hotter or colder than the material of the calorimeter. The initial and final temperatures of the water, t_1 and t_2, and the weight of water W should be observed. If Q is the water equivalent, and t is the initial temperature of the calorimeter, then

$$W(t_1-t_2)=Q(t_2-t)$$

since the heat lost by the water equals that gained by the calorimeter, or vice versa. This equation may then be solved for Q.

The equation neglects radiation. To avoid the consequent error, a second test may be made with the temperatures of calorimeter and water so adjusted that the temperature rise up to room temperature approximately equals that above.

For example, suppose that for the first test water at 110° F. cools to 80°, raising the temperature of the calorimeter from that of the room, 70°, to 80°. The rise of temperature of the calorimeter during the test is 10°. Now, if its initial temperature were 65° and its final 75°, radiation to and from the room would be balanced. Hence, for the second test, the calorimeter should be previously cooled to 65°, and then heated with water at 105° so that the 30° drop in the water temperature, noted in the first test, will bring the final temperature to the desired 75°. The variation will not, of course, be exactly equal above and below the room temperature, but by this method the effect of radiation upon the accuracy of the determination of the water equivalent is made negligible.

The student should note the fact that radiation thus taken care of is an entirely different quantity from that which occurs

in the usual operation of the instrument. For accurate work, the latter should be taken into account as follows.

(b) Determination of the Radiation Correction. For non-continuous condensing calorimeters such as the barrel calorimeter, radiation correction may be avoided in actual use in exactly the same way as that described for the water equivalent determination. The flow of steam into the calorimeter is continued sufficiently long to raise the temperature of the cold condensing water as much above room temperature as it was below.

For continuous calorimeters, the radiation correction must be found and applied in the equation, as follows,

$$\begin{aligned} W_z\{(h+xL)-(T_s-32)\} &= W_w(T_w-t_w)+R \\ &= W_w(T_w+t'-t_w), \end{aligned}$$

in which R is the heat radiated from W_w lbs. of cooling water and $t'=R\div W_w$.

To find t', the calorimeter is operated as in usual practice, except that dry steam is furnished to it either by the method of Test 32 (*a*) or by using a steam separator. The value of x is then 1, and the equation may be solved for t'.

The correction applies only to the temperature conditions of the radiation test. If the conditions in ordinary operation are markedly different, the radiation test should be repeated to fit the new conditions. It is possible, however, to keep the initial and final temperatures of the cooling water practically the same for different conditions of the steam samples by varying the rate of cooling water. If this is done, and if the room temperature is not very variable, only a single value of the radiation correction is needed.

Problem 34_1. The data from a barrel calorimeter determination are as follows: Weight of water in barrel before condensing = 350 lbs., temperature = 55° F. Weight after condensing = 370.6 lbs., temperature = 115°. Pressure

of steam sample = 100 lbs. gage. What is the quality, neglecting the water equivalent and radiation? *Ans.*, 0.901.

Problem 34_2. A water equivalent determination for the preceding example shows that 300 lbs. of water will fall from 98.6° to 97.3° F. when placed in the barrel at 60°. What is the water equivalent, and what is the corrected value of the quality? *Ans.*, 10 lbs.; 0.934.

Problem 34_3. The circulating water in a surface condenser rises in temperature from 50° to 86° F., 275 lbs. being used to condense 10 lbs. of a steam sample at 80 lbs. gage. The temperature of the condensate is 98°. What is the quality of the sample? *Ans.*, 0.856.

FRICTION

Friction is the force which resists the relative motion of two bodies in contact, and is due to the interference of their particles at the surface of contact. The laws controlling this force are different depending upon whether the rubbing materials are fluid or solid. Solid friction may be either from sliding or rolling contact, but in this work sliding friction only will be considered. There is a third case, generally of greatest importance to the mechanical engineer, namely, the friction of lubricated solids. The controlling laws fall between those for solid and fluid friction, resembling the one or the other according to the amount of lubrication.

The laws of fluid friction are well established, but this is unfortunately not true of solid friction. An understanding of the latter is important in the analyses of friction drives, belt forces, etc., and important also to our knowledge of lubricated friction since the rubbing action of solids is a limiting condition of lubricated machine parts.

It has generally been assumed that the friction between two dry solids is a constant fraction of the normal force pressing the solids together, independent of the area of contact and velocity of sliding, and depending only upon the nature of the surfaces. These hypotheses have been discredited, but, thus far, experi-

mentation upon this subject has not proceeded enough to give us more than an idea wherein they are fallacious. It is possible that they are true for certain materials or certain conditions, but unquestionably they are not generally true.

The variation of friction, in the general case, depends upon the normal force between the surfaces, the amount of contact area, the velocity of sliding, and the materials of the bodies.

The following table is arranged to give a ready comparison of the several cases. It should be remembered, however, that the statements under solid and lubricated friction are not well established, and are possibly subject to exceptions.

VARIATION OF FRICTION

	Fluid.	Solid.	Lubricated Solid.
Contact area.	Directly proportional.	Independent (approx.)	Directly proportional.
Velocity.	Directly proportional as the square, except at low speeds.	Inversely, at a diminishing rate.	Inversely for low speeds, directly as square root for high speeds.
Character of surface.	Independent.	Varies.	Varies.
Normal force.	Independent.	Directly proportional.	Independent.

It is further known that the frictional resistance to a fluid moving upon a solid is related in some way to the viscosity of the fluid, decreasing as the viscosity decreases. This property has an important bearing upon lubricated solid friction because the viscosity of lubricants varies with temperature, thus introducing another variable in this case.

For solid friction, the working hypothesis is that

$$F = fN$$

in which F is the force of friction, N the force normal to the rubbing surfaces, and f a number less than one called the "coefficient of friction."

35. Oil Testers

Principles. Oil testers are instruments for determining the coefficient of friction in the case of lubricated solids. Their operation generally depends upon the balancing of the friction by a static moment which can be measured; from this and the constants of the instrument, the coefficient may be calculated. Referring to Fig. 73, S is a revolving shaft to which is fitted a bearing carrying a pendulum. It is arranged that the bearing swing freely and bear on the shaft with a measured force, N. The oil to be tested is fed to the bearing, and the shaft caused to revolve in a clockwise direction. The force of friction then swings the pendulum to such an angle as to establish equilibrium. If the radius of the shaft is r ft., then the moment of the frictional force, fN, is $fN \times r$. The weight W of the pendulum acts on an arm of $R \sin A$, R being the distance of the center of the weight to the center of the shaft; consequently the moment of the pendulum equals $W \times R \sin A$. As this moment equals the moment of the frictional force, we have

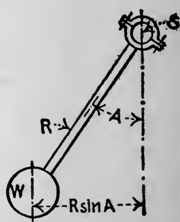


Fig. 73.
Oil Tester.

$$fNr = WR \sin A,$$

and

$$f = \frac{WR \sin A}{Nr}.$$

For a given set of conditions, all of the quantities on the right

of the equation are constant except $\sin A$, which thus becomes a measure of f.

The Thurston oil testing machine depends upon this principle. The force N is applied and varied by means of a spring mounted on the pendulum, by tightening which the two parts of the bearing may be clamped more or less tightly together. The total force, N, equals twice the tension of the spring plus the weight of the pendulum. There are a pointer and scale on the pendulum by which the total force may be read directly. Another pointer on the pendulum moves over a stationary scale graduated to values of $WR \sin A \div r$. A thermometer well is provided in the upper bearing so running temperatures may be ascertained.

The Riehlé oil testing machine carries a hand screw for applying the pressure, and the pressure is measured by means of a scale beam. A second scale beam measures the moment of the turning force induced by friction. By equating this moment to fNr, as in the Thurston machine, the coefficient of friction may be obtained.

(a) Constants of the Instrument. In the Riehlé machine, the only constant generally necessary to test is the value of r. If desired the indications of the beams may be tested according to the principles of Test No. 1.

With the Thurston machine, it is necessary to find WR, r, and to test the spring.

To find WR, the pendulum should be swung to a horizontal position and supported on a pedestal resting on a platform scales. A test is then made similar to that for the unbalanced weight of a Prony brake, Test No. 6 (*a*). The result should be multiplied by the horizontal distance between the point of application of the pendulum on the pedestal and the center of motion of the pendulum. This gives the desired moment, WR.

In addition to measuring the radius of the bearing, r, its length should be determined, so that the bearing pressure in pounds per square inch of projected area may be calculated.

The graduations on the stationary scale may be tested with these data by calculating their values at any values of the angle A.

To test the indications of the spring its scale should be determined according to Test No. 2. As the spring is a heavy one, a strength testing machine is required to compress it.

Problem 35₁. Run a series of tests with an oil testing machine to show the relation between coefficient and load in pounds per square inch.

Problem 35₂. Run a series to show the relation between coefficient and velocity.

Problem 35₃. Run a series to show the relation between coefficient and temperature.

36. Belt Testers

Principles. When power is transmitted from one shaft to another by means of a belt running on pulleys, friction is the useful force. Friction sets up tensions in the belt; on one side, a greater amount T_1, and on the other a lesser, T_2 (see Fig. 74). The difference between these tensions is the net force, T, which effects a turning effort of the follower. It is shown in works on machine design that

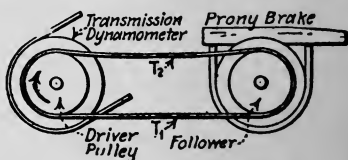


Fig. 74.—Belt Tester.

$$\frac{T_1}{T_2}=10^k \quad \text{and} \quad T=T_1-T_2=T_1\left(1-\frac{1}{10^k}\right)$$

in which $k=.0076f\theta$; f being coefficient of friction and θ the angle of contact of the belt in degrees. The formulas are derived on the assumption that f varies only with the normal force of contact and that centrifugal force on the belt is negligible. For belt speeds over 2500 ft. per min., centrifugal force must be

considered. Its effect is to lessen the normal force between belt and pulley, thus decreasing the effective force, and

$$T = T_1 - T_2 = \left(T_1 - \frac{12wv^2}{g}\right)\left(1 - \frac{1}{10^k}\right),$$

in which w is the weight in pounds of a strip of belt 1 in. long and full width; v, its velocity in feet per second; and g, the acceleration of gravity.

In usual operation, there is always a certain amount of "slip" between the belt and the pulleys. Instead of running at the same linear velocity of points of contact, there is a relative velocity difference between the belt and each pulley rim. This fact invalidates to some extent the formula giving the relation between the belt tensions, since friction varies with the velocity of rubbing. Furthermore, so little is definitely known about solid friction, that the assumption $F = fN$ is of doubtful accuracy. It is very likely that, in the case of belts, the coefficient of friction f varies with the normal force N as well as with the velocity of slipping. As the formula depends upon an opposite assumption, results of the coefficient of friction obtained from it must be considered of very questionable value. On the other hand, we have no better formula, and for this reason it must be used.

Belt-testing machines are instruments for testing the efficiency of transmission, for determining the slip under given conditions, and for determining the coefficient of friction.

The efficiency of transmission may be found by measuring the power delivered to the belt with a transmission dynamometer and the power delivered to the follower pulley by an absorption dynamometer (see Fig. 74), suitable allowance being made for the friction of the shaft bearings. The difference between these quantities is the losses, which are due to work done by friction in moving through a distance equal to the slip and the work done in flexing the belt, windage neglected.

The slip may be ascertained by counting the R.p.m. of the shafts; the difference between the linear velocities of the pulley rims is the total slip. The slip over either pulley may be found independently by measuring the linear velocity of the belt. This may be done by counting the number of times a chalk mark on the belt appears in a given time or by arranging a piece of metal to project from the edge of the belt so as to trip a revolution counter. Slip should be expressed in per cent of the driving pulley rim speed. Some forms of belt-testing machines have a differential gear arrangement by which the per cent of slip may be directly read.

To test for the coefficient of friction, apparatus must be provided for the determination of the belt tensions, T_1 and T_2, so that the formula may be solved for f. This is usually done by separate measurements of the sum and difference of the belt tensions; adding these results and dividing by two gives T_1, knowing which, T_2 is readily obtained.

The turning force of the belt, T, equals the difference of the tensions. The moment of this turning force equals the torque shown by the friction brake (Fig. 74). Therefore, the difference of belt tensions may be calculated by dividing the torque shown by the brake by the radius of the follower pulley. This radius should be measured to the center line of the belt.

To get the sum of the belt tensions, not including that due to centrifugal force, one type of machine has the driver pulley mounted freely on one arm of a bell-crank lever the other arm of which bears on a platform scales. The belt pull is then indicated on the scales in inverse proportion to the lever arms. Another type of machine has the belt arranged vertically, the follower pulley being freely suspended. The sum of the belt tensions is then equal to the weight of the follower pulley and attachments together with any additional weights used for the purpose of increasing the belt tensions. In both types, the total tension may be varied.

The coefficient varies markedly with the condition of the rubbing surfaces, so every precaution should be used to keep this and other controllable conditions constant. The coefficient also varies with the humidity of the surrounding atmosphere; it is therefore well to make observations of this.

(a) **The Constants of the Instrument** are those of the Prony brake (see Test No. 6) and of any lever arms used for getting the tensions. These are readily obtainable by direct measurement.

Problem 36_1. The torque as shown by the friction brake is 100 lb.-ft. If the follower pulley is 2 ft. in diameter, what is the difference between the belt tensions? If the sum of the belt tensions is 200 lbs. and the angle of contact is 190°, what is the coefficient of friction? *Ans.*, $f=0.33$.

Problem 36_2. Describe how a test should be made to show the variation of the coefficient with slip, all other conditions being constant.

Problem 36_3. Describe how a test should be made to show the variation of the coefficient with the normal force of contact, slip and other conditions being constant.

PART TWO

THE ANALYSIS OF COMBUSTION

THE CONSTITUENTS OF FUELS

THE principal commercial fuels are coals, oils, and gases in their various forms. The elemental constituents of these fuels, which for the most part are common to all of them, are carbon, hydrogen, oxygen, nitrogen, and sulphur. Of these constituents the ones depended upon to yield heat through combustion are carbon and hydrogen. Sulphur has a heat value, but it is an undesirable element in fuel since in coal it goes to form clinker and in gas it may combine with water to make sulphuric acid, these processes accompanying the combustion of the fuel. Nitrogen and carbon in the form of CO_2 are inert gases and are valueless to combustion.

Carbon and hydrogen occur free or in combination with each other as hydrocarbons or in combination with oxygen. The only combustible carbon-oxygen compound is carbon monoxide, which is one of the most important of fuel gas constituents.

Coal may be classified broadly as anthracite, semi-anthracite, semi-bituminous and bituminous, distinguishable by the amounts of so-called "volatile matter" they contain. This is burnable material, mainly hydrocarbons, which may be driven from the coal without ignition. The usual percentages by weight of volatile matter in the "combustible" of the four classes of coal are given in the following table, which also shows the amounts

that are left of carbon (referred to as "fixed carbon") when the volatile matter is driven off. By "combustible" is meant the volatile matter and fixed carbon, leaving out of consideration ash and moisture.

PROXIMATE ANALYSES OF DRY COMBUSTIBLE

	Anthracite.	Semi-anthra.	Semi-bit.	Bituminous.
Volatile matter.........	3 to 7.5	7.5 to 12.5	12.5 to 25	25 to 50
Volatile matter, average.	5	10	19	37.5
Fixed carbon...........	95	90	81	62.5

The ash from coals is the residue when the combustible parts have been burned, and consists of such materials as clay, sand, slate, etc. The ash does not vary with the class of the coal, and may be widely different in proportion and quality in each of the classes. Water is also contained in coals, the amount varying with atmospheric and storage conditions.

It should be observed that anthracite coal is mainly carbon and ash, there being generally very little volatile matter.

A complete chemical analysis of a coal is called its "ultimate analysis." The following table is for representative samples of the American product, and gives percentages based on the combustible.

ULTIMATE ANALYSES OF COALS

	Anthracite.	Semi-anthra.	Semi-bit.	Bituminous.
Carbon..............	95	90.5	87	81
Hydrogen............	2.0	5	5	6
Nitrogen.............	0.8	....	1.5	1.5
Oxygen..............	2.0	4	4	7.5

When such an analysis is required it should be obtained from a chemist, as the methods and apparatus are remote from mechanical

experimentation. An analysis yielding the percentages of moisture, volatile matter, fixed carbon and ash, however, may readily be made by the skilled engineer, and often serves all the purposes in the study of combustion. This analysis is called the "proximate analysis."

The kind of coal and its properties as fuel are determined roughly by the locality at which it is mined, but it should be noted that they may vary markedly in samples even from the same mine.

The commercial grades of coal with relation to size are as follows, being listed in the order of the size, largest first.

ANTHRACITE	SIZE, INS.	BITUMINOUS
Broken	4	Run of mine
Egg	3	Lump, various sizes
Stove	2	Nut, various sizes
Chestnut	$1\frac{1}{4}$	Screenings, various sizes
Pea	$\frac{7}{8}$	Washed sizes
Buckwheat, No. 1	$\frac{7}{16}$	
Buckwheat, No. 2	$\frac{1}{4}$	
Buckwheat, No. 3	$\frac{1}{8}$	

Oils used for fuel are composed of different hydrocarbons of the form C_nH_{2n+2} or C_nH_n, which have a wide range in such physical properties as specific gravity, volatility, burning and flash points. Gasoline and kerosene are the lighter distillates from crude oil.

The percentages by weight of carbon and hydrogen in various fuel oils and their distillates from all parts of the world are not very different. Carbon is generally between 83 and 87 per cent and hydrogen between 11 and 15 per cent, the rest being oxygen, nitrogen and sulphur.

Gases. The principal gases used for fuel are illuminating gas, producer gas, natural gas, and blast furnace gas. The

following table gives the percentages by volume of the constituents of representative American gases.

CONSTITUENTS OF GAS FUELS FOR POWER

"Low" means less than 1 per cent.

	Producer.	Illuminating.	Natural.
Carbon monoxide, CO	25	15	low
Hydrogen, H_2	12	45	2
Methane, CH_4	2	25	95
Ethylene, C_2H_4; Benzol, C_6H_6; Heavy hydrocarbons	low	5	low
Oxygen, O_2	low	1	low
Sulphureted hydrogen, H_2S; Sulphur dioxide, SO_2	low	low	low
Carbon dioxide, CO_2	5	4	low
Nitrogen, N_2	55	5	2

Blast furnace gas is combustible mainly through CO. It contains a little hydrogen and methane, and is high in nitrogen.

Analysis of these gases must be made by a chemist, with the exception, possibly, of producer gas and blast furnace gas. These may be analyzed by the Orsat apparatus (Test No. 40) if a pipette for the determination of hydrogen is provided, methane being ignored.

37. The Proximate Analysis of Coal

Sampling. When an analysis of a small sample is made to represent a large amount of coal, every precaution should be taken to secure a truly representative sample. For the purposes of boiler and producer tests, this may be done as follows.

As each barrowful or charge of coal is removed from the pile for firing, an amount varying between a handful and a shovelful is withdrawn and put in a closed receptacle to make up a gross sample. The quantity of coal in each of these samplings and of

the gross sample necessary for accurate results depends upon the lump size of the coal, upon its homogeneity, and to a limited extent upon the total amount used. For the smallest sizes of coal of fairly uniform quality, the gross sample may be no more than 100 lbs.; under other conditions, it need not exceed 250 lbs. Assuming 200 lbs., the gross sample is broken on a clean surface, preferably a piece of sheet iron, with the assistance of a tamping bar or weight, so that the largest lumps are not more than $\frac{1}{2}$ in. diameter. This coal is then mixed and piled in the form of a cone. The cone should be "quartered" into four heaps by passing a board through the cone axis in two planes at right angles. Two diagonally opposite heaps are then discarded and the remaining two mixed, broken to $\frac{1}{4}$ in., coned and again quartered. The process of quartering and discarding should be repeated until about 5 lbs. of coal lumps not larger than $\frac{1}{4}$ inch are left. Half of this may be put aside in a sealed glass jar for check determinations if desired; the balance is to be run through a coffee mill or other crusher adjusted so that the lumps are reduced to about $\frac{1}{16}$ in. and less. After this has been thoroughly mixed, a sufficient quantity of it for the proximate analysis is crushed by hand with mortar and pestle until all of that quantity is of such lump size as to pass through a sieve of 20 meshes to the inch; this is put in a corked bottle and used as follows.

(a) **Determination of Moisture, Volatile Matter, fixed Carbon and Ash.** About 4 gms. of the 20-mesh sample are carefully weighed. It is then spread out on a large watch-glass, and placed in a gas oven where it is kept at a temperature of 220° to 230° F. for one hour. Next it is placed in a desiccator to cool so that no moisture may be absorbed from the air. When cool enough to weigh accurately, it is covered, removed from the desiccator and weighed.

The difference between the first weight and the last is the weight of moisture. The percentage is figured on the basis of the weight of the sample including moisture.

The gross sample may lose some of its surface moisture during the processes of quartering and crushing, by evaporation to

the air. To avoid this as much as possible, reduction of the gross sample should be performed in a cool room in which the atmosphere is neither very dry nor very moist, and it should be accomplished as quickly as possible. Some experimenters prefer to find moisture from a sample of the $\frac{1}{4}''$ size weighing about 5 lbs. and dried for 24 hours.

To determine volatile matter, about 1 gm. of the 20-mesh coal, pulverized to pass through a 60-mesh sieve, is placed in a covered crucible of previously found weight and put in the hottest part of the flame of a Bunsen burner which is 8 ins. in height when burning free. After seven minutes, it is removed, cooled in a desiccator, and weighed. The difference in weight of the sample before and after heating equals the sum of its moisture and volatile matter. Knowing the former, the latter may be readily calculated.

In this determination, it is important to exclude air from the crucible as otherwise some of the fixed carbon may be driven off as CO_2. Some experimenters prefer to wet the coal after the first weighing so that the steam generated will displace the air in the crucible before the distillation of the volatile matter.

Fixed carbon is found by subjecting the residue from the last test to the heat of a blast flame, the cover of the crucible being removed. Occasional stirring with a platinum wire assists combustion. It is generally a lengthy process to consume all of the fixed carbon; it may be hastened by directing a gentle stream of oxygen from a cylinder filled with that gas into the crucible. When the carbon is all burned, its amount is ascertained by cooling the sample in the desiccator, and weighing as before.

The ash is determined by weighing the residue from the last test.

Sulphur, which occurs both in the volatile matter and in the ash, is sometimes separately analyzed for in the proximate analysis.

To shorten the total time necessary for the proximate analysis, it is recommended that the tests for volatile matter, fixed carbon, and ash be started during the time that a separate sample is being dried out for the moisture determination. These tests will show

volatile matter and fixed carbon plus moisture; allowance for the latter may be made afterwards.

(b) Calculation of Hydrogen and Volatile and Total Carbon from the Proximate Analysis. For the complete analysis of boiler and producer performance, it is necessary to know the percentage of total carbon in the fuel and of hydrogen in the volatile matter. As the proximate analysis shows only the fixed carbon, that in the volatile matter may be estimated and added to the amount of fixed carbon to get the total. Professor Lionel S. Marks provides an empirical method of doing this and of estimating the hydrogen in the volatile matter. He has pointed out a relation applying to American coals by means of curves,* and these curves have been expressed, in part, by the following equations due to Professor Diederichs:

Let h_c = hydrogen, exclusive of that in moisture;
c_c = carbon in the volatile matter or "volatile carbon";
v_c = volatile matter:

all expressed as weight-percentages of the combustible. Then

$$h_c = v_c\left(\frac{7.35}{v_c+10} - 0.013\right);$$

$$c_c = 0.02v_c^2 \quad \text{or} \quad 0.9(v_c - 10) \text{ for anthracites;}$$

$$c_c = 0.9(v_c - 14) \text{ for bituminous coals.}$$

As an example of the use of these formulas, take the proximate analysis given on page 270, and let it be required to find the hydrogen in one pound of the coal, H_t, and the total carbon, C_t.

The combustible, being the sum of the fixed carbon and volatile matter, is .807 + .0617 = 0.869 lb. per lb. of coal.

* *Power*, Dec., 1908.

Hence

$$v_c = \frac{.0617}{.869} \times 100 = 7.1 \text{ per cent,}$$

and

$$h_c = 7.1\left(\frac{7.35}{7.1+10} - 0.013\right) = 3 \text{ per cent.}$$

This is the percentage of *hydrogen based on the combustible.* To base it on the coal, we have

$$H_t = \tfrac{3}{100} \times 0.869 = .026 \text{ lb. of hydrogen per lb. of coal.}$$

Similarly for the volatile carbon,

$$c_c = .02 \times 7.1^2 = 1 \text{ per cent.}$$

Consequently in 1 lb. of coal, there is $\tfrac{1}{100} \times .869 = .00869$ lb. of volatile carbon, and the total carbon is

$$C_t = .807 + .00869 = .816 \text{ lb.}$$

Problem 37$_1$. The proximate analysis of a coal gives moisture = 1 per cent, volatile matter = 22, fixed carbon = 72, and ash = 5. What are the percentages of hydrogen and carbon in the volatile matter, and what is the percentage of total carbon? Base answers on coal as analyzed, not on combustible. *Ans.*, 4.6, 8.5, 80.5%.

Problem 37$_2$. Give the proximate analysis in the preceding problem on the basis of "dry" coal instead of coal "as received." *Ans.*, 1.01%, etc.

THE HEAT VALUE OF FUELS

When used for the generation of power, combustion may be defined as the rapid chemical combination of the oxygen in air with the combustible constituents of fuel, which combination is accompanied by the evolution of heat. It has been pointed out that the elemental combustibles in fuels are carbon, hydrogen, and to a lesser degree, sulphur. The complete combustion of carbon forms carbon dioxide, CO_2, and hydrogen, water, H_2O.

When a unit mass of fuel is burned completely, the heat evolved raises the temperature of the materials entering into the combination and of surrounding objects. If the products of combustion are cooled to the temperature before combustion, then the total heat given up is the "heat" or "calorific value" of the fuel. Or, more briefly, *the heat value of a fuel is the number of heat units that are released by the complete combustion of a unit mass of the fuel.*

It should be noted that to release all of the heat generated, the products must be cooled down to room temperature. This may be done in two ways, namely, so that the H_2O formed by combustion of the hydrogen remains as steam or so that the H_2O is condensed. In the former case the latent heat of the steam remains in the products of combustion, the heat released is correspondingly less, and is referred to as the "lower heat value." In the latter case the latent heat of the steam is included in the heat released which is then called the "higher heat value."

Whether the one or the other quantity should be used depends upon the character of the test for which the heat value is needed.

The difference between the two values depends upon the amount of hydrogen in the fuel. For coals, it is comparatively small, but for gases and oils it may be as high as 15 per cent.

The units in which heat values are expressed in the United States are as follows, corresponding French units being used to a limited extent only.

For solids, B.t.u.s per pound.
For liquids, B.t.u.s per pound or per gallon.
For gases, B.t.u.s per standard cubic foot.

The standard cubic foot of gas is a cubic foot of gas under standard conditions of pressure and temperature. That this specification is necessary will be seen when it is considered that the mass of gas in a cubic foot depends upon these conditions, and consequently its heat value. The standard conditions of pressure and temperature are either 29.92 ins. of mercury and 32° F., or 30 ins. of mercury and 62° F. These pressures

are the same and equal to 14.7 lbs. per square inch, the difference in the mercury columns being due to the temperature differences.

In this work the 32 degrees standard will be used.

The following gives experimentally determined heat values of the elemental combustibles and of various gases, and of fuels.

HEAT VALUES IN B.T.U. PER POUND

Carbon burned to CO_2	14,600
Carbon burned to CO	4,400
Sulphur burned to SO_2	4,000
Hydrogen to H_2O	62,000, higher 52,500, lower

HEAT VALUES IN B.T.U. PER STANDARD CUBIC FOOT (32°)

	Higher.	Lower.
Carbon monoxide, CO	342	....
Hydrogen, H_2	346	294
Methane, CH_4	1065	955
Ethylene, C_2H_4	1680	1560
Benzol, C_6H_6	4000	3830

HIGHER HEAT VALUES OF COMMERCIAL FUELS, B.T.U.

Coals	11,000 to 15,000 (per lb.)
Oils	18,000 to 20,000 "
Illuminating gas, average	550 (per cu. ft.)
Producer gas "	150 " "
Natural gas "	1050 " "

If it is desired to refer the heat values of the gases to the 62° standard, it is only necessary to multiply by 0.943, the ratio of absolute temperatures.

Instruments for determining the heat value of fuels are called "fuel calorimeters."

38. The Determination of the Heat Value of Coal

Principles. This may be done by calculation from empirical formulas involving the results of the proximate or ultimate analysis of the coal, or by the use of a fuel calorimeter. The latter method is the more accurate.

Results from coal calorimeters are obtained by burning a small sample of the coal in an air-tight chamber immersed in water. The heat given up to the water and calorimeter parts by the combustion of this coal is measured by the rise of temperature, from which the heat value of the coal may be calculated.

The essential parts of a coal calorimeter are the water vessel, jacketed to prevent radiation, the combustion chamber or "bomb" with a device for igniting the charge, and a thermometer or its equivalent. Ignition is accomplished either by an electric current or by dropping a bit of white hot wire through a tube into the bomb, a valve being arranged to allow its entrance and to prevent the exit of the gases generated. To support combustion, oxygen is charged with the coal, either in the free state from a cylinder under pressure, or in the form of a chemical rich in oxygen.

The Emerson and the Mahler are examples of calorimeters with which free oxygen is used for the combustion of the fuel. With the Parr calorimeter sodium peroxide is used as a source of oxygen.

Sampling of the coal to be tested should be done exactly as for Test 37, except that the last crushing should be to particles that will pass through a 100-mesh sieve. The final sample should be dried in an oven at 220° to 230° F. before it is charged in the calorimeter.

(a) **Determination of Heat Value by the Emerson or Mahler Calorimeter.** Let W stand for the weight of calorimeter water in grams, and w the water equivalent of the bomb, water vessel and other parts in contact with the calorimeter water (that is, the weight of water that would absorb the same amount of heat as these parts); and T the temperature, deg. C., rise brought about by the combustion of C grams of coal. The heat given up by the combustion is then $(W+w)\times T$ calories, and the heat value of the coal is $(W+w)T\div C$ calories per gram. To convert this into B.t.u. per pound, multiply by 1.8. The temperature rise, T, as observed, must be corrected as will be described. More detailed instructions for operation follow:

A small test tube containing about 2 gms. of the pulverized dry

coal is weighed on a chemical balance to an accuracy of one milligram. About 1 gm. of this coal is poured into the pan provided for the bomb charge (not less than 0.8 gm. or more than 1.2 gm.). The test tube with the remaining coal is then weighed again, and the weight of the bomb charge obtained by difference. After the ignition wire is adjusted the bomb cap should be screwed in place, a little vaseline on the threads enabling a tighter fit. Care must be taken that no vaseline be in the combustion space. The bomb is now charged with oxygen at 300 lbs. per sq. in., and then immersed in the calorimeter, the water for which has been previously measured, and heated (if necessary) to room temperature or a trifle above. The jacket water temperature, too, should be closely that of the room.

A small motor-driven paddle is provided to circulate the water in the calorimeter vessel. Having started this with everything in place, readings of temperature from a thermometer graduated to 1-100 deg. C. should be made each minute for five minutes. Ignition is accomplished by throwing current through the fuse wire at the end of the fifth minute. The temperature should now be read each half-minute until the maximum temperature is reached, and thereafter each minute for five minutes. A curve of temperature vs. time is needed.

The temperature readings before ignition and after combustion are obtained in order to find the radiation correction. An accurate method for calculating the radiation is that of Pfaundler, for which see White's "Gas and Fuel Analysis." Peabody's method is much shorter and is sufficiently accurate when the time from the instant of ignition to that of maximum temperature is about one minute. Peabody's method is as follows:

From the temperature-time curve find

R_a = no. of degrees radiation per half minute interval before ignition,

R_b = no. of degrees radiation per half minute interval after maximum temperature is passed,

N = no. of half minute intervals from instant of ignition to instant of maximum temperature.

Then the correction for radiation is $R_a+(N-1)R_b$.

Note that in this expression R_a or R_b should be taken as plus when the temperature is falling and minus when rising.

The final radiation is always plus (that is from the calorimeter to the room); therefore, if the temperature before ignition was at or a trifle above that of the room, the correction must be added to the observed rise.

For measuring the actual temperature changes, a Beckmann differential thermometer (graduated from 0 to about 5 deg. C.) is desirable. This instrument has a mercury container at the upper end of the capillary tube as well as at the lower; the upper acting as a reservoir for any mercury not needed in the lower, or bulb. If the column of mercury in the capillary stands too much above the zero graduation when the bulb is immersed in the calorimeter water, some of it may be forced, by gently heating the bulb, into the upper reservoir. Provision is made so that this surplus of mercury can be segregated from the main column and confined in the upper reservoir, whereupon the main column will stand at a lower point when cooled to the calorimeter temperature. Two or three manipulations of this sort are necessary to make the column stand at the desired graduation, about 1 deg. C., for the initial reading.

If the mercury stands below the zero graduation, upon the first immersion in the calorimeter water, the deficiency of mercury must be supplied from the upper reservoir by a reverse operation.

For extremely accurate work, other corrections than for radiation may be made. These include the heat of fusion of the ignition wire, and the heat due to the formation of nitric and sulphuric acids due to the use of oxygen instead of air for combustion. These corrections, however, may be omitted for ordinary work.

The water equivalent of the calorimeter may be found by using the corrected temperature rise when 1 gram of pure naphthaline

is burned. Naphthaline has a known heat value of 9610 calories per gram, which enables a calculation of w in the formula: Heat value $= (W \div w)T \div C$.

(b) Determination of Heat Value by the Parr Calorimeter. Let W stand for the weight of water used in pounds, w, the water equivalent of the bomb, water vessel and parts (that is, the weight of water that would absorb the same amount of heat in the same temperature range), and T, the temperature rise brought about by the combustion of C lbs. of coal mixed with the appropriate weight of sodium peroxide. The heat given up by the combustion is then $(W+w)T$. Now, 27 per cent of this heat is due to the by-reactions of the sodium peroxide, so that there remains an amount equal to 0.73 $(W+w)T$ generated by the coal. Therefore, the heat value of the coal is

$$\frac{.73(W+w)T}{C}.$$

The water equivalent, w, may be taken as 0.3. Generally 2 liters of water and $\frac{1}{2}$ gram of coal are used. In English units, these weights are 4.41 and 0.001102 lb. Combining the constants, we have

$$\text{Heat value} = \frac{.73(4.41+0.3)}{.001102} T = 3117T.$$

The temperature T is the observed temperature minus certain corrections. The hot wire adds some heat which should be allowed for by subtracting 0.022° F., from the observed temperature. Anthracite coal is somewhat difficult to burn completely in the bomb, so an additional chemical (about 1 gm.) called "accelerator" is used, the heat from which should be allowed for. The accelerator consists of 2 parts by weight of potassium persulphate and one of ammonium persulphate. The corrections are 0.005 degree for each per cent of ash and 0.010 for each per cent of sulphur, to be subtracted.

Radiation from the calorimeter is very small. It may be corrected for roughly by noting the fall in temperature immediately after the maximum temperature has been reached and during an interval of time equal to that between ignition and the occurrence of maximum temperature. The correction should be added to the observed rise.

Radiation may be practically obviated by making the initial temperature of the water about 2° less than that of the room. As the observed rise in temperature is between 4 and 5° the radiation will be balanced.

The procedure for the test is to add about 9 gms. of sodium peroxide to $\frac{1}{2}$ gm. of coal, mix thoroughly in the bomb, ignite, and, during the period of heating, to revolve the bomb in the calorimeter so as to circulate the water. This last is accomplished by means of vanes attached to the bomb, the purpose being to quickly make the temperature of the water uniform. The temperature rise is noted from a finely graduated thermometer.

Care should be taken not to allow water to touch the sodium peroxide, as they react violently.

(c) **Calculation of Heat Value from the Fuel Analysis.** If the combustibles in coal were in the elemental form, it would be easy to calculate the heating effect from each according to its amount and thus get the heating value of the coal. This, however, is not so; they are combined as hydrocarbons and otherwise. Now when combustion takes place some heat must be given up to the hydrocarbons to separate the carbon from the hydrogen so that they may recombine with oxygen. That is, some heat becomes latent through dissociation of the elements which lessens the heat available from their combination with oxygen. So there is an interchange of heat in burning coal which makes its heat value dependent upon the previous combination of its elements. We therefore cannot calculate heat values with precision in this way, but certain empirical for-

mulas enable us to make fair estimates. One of them is Dulong's as below.

$$\text{Heat value} = 14{,}600C + 62{,}000\left(H - \frac{O}{8}\right) + 4000S,$$

the symbols C, H, O, and S standing for the weights in pounds of carbon, hydrogen, oxygen, and sulphur in 1 lb. of the fuel, respectively. The result is in B.t.u. per pound. This formula uses the ultimate analysis of the coal, and is more applicable to anthracite than to bituminous coals.

Another formula, using the proximate analysis, is Goutal's.* This is

$$\text{Heat value} = 147.6 \times fc + K \times vm,$$

in which fc and vm are the percentages of fixed carbon and volatile matter in the fuel as received, respectively, and K has different values depending upon the amount of volatile matter, as shown in the curve, Fig. 75.

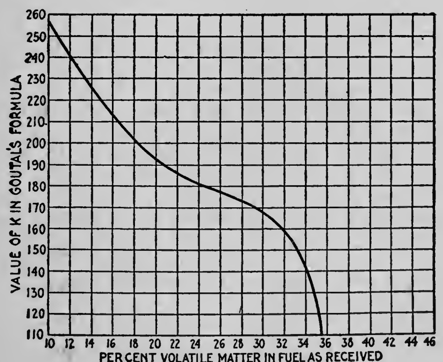


FIG. 75.

Fig. 76 is a reproduction of Mahler's curve, from which, probably, may be had as reliable a result of heat value as any

* *Wisconsin Engineer*, Dec., 1911.

of the relations proposed for this purpose. As an example of its use assume a coal with 16.9 per cent of volatile matter and 70.8 per cent of fixed carbon. Then

Combustible in the coal $= 16.9 + 70.8 = 87.7$ per cent;
Fixed carbon in the combustible $= 70.8 \div 87.7 = 80.6$ per cent.

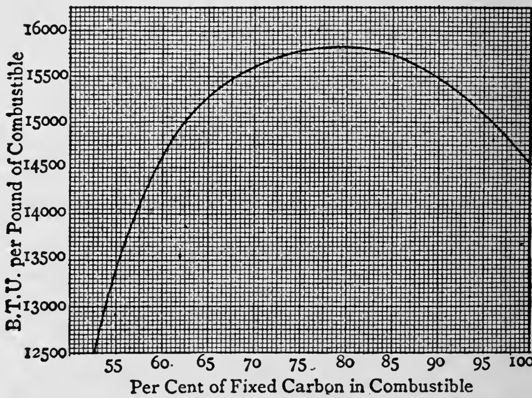


FIG. 76.—Mahler's Curve for Coal Heat Values.

From Fig. 76, against 80.6 per cent is found the heat value of the combustible, 15,800 B.t.u. per pound. Consequently, the

Heat value of the coal $= 15{,}800 \times .877 = 13{,}900$ B.t.u.

It should be emphasized that all of these relations for calculating the heat value of coal are approximate, and generally give better values with the lower percentages of volatile matter; the probable error then being within 2 per cent.

Problem 38$_1$. A sample of bituminous coal is tested for heat value. Ignition by hot wire. 2.5 liters of water used. Water equivalent of calorimeter $= 0.40$ lb. Sample weighs 0.6 gm. How much heat is generated and what is the heat value if temperature rise is 4.23° F.? *Ans.*, 13,300 B.t.u.

Problem 38$_2$. Calculate the heat value of coal giving a proximate analysis as follows: moisture, 2.75; volatile matter, 6.00; fixed carbon, 78.45. Ash, 12.8 per cent. Use Mahler's curve. What would be the heat value if the coal were dry? *Ans.*, 12,900 and 13,300 B.t.u.

39. The Determination of the Heat Value of Gases and Oils

Principles. The heat values of gases may be calculated readily from their chemical analyses, but this is not true of oils on account of their complex structure. Both oils and gases may be tested for heat value by the use of a properly designed calorimeter. The Junker calorimeter is universally used for this purpose.

This instrument is an ingenious arrangement of heating surfaces surrounded by water, by which all the heat generated by the fuel to be tested is passed to the water. The water is kept flowing through the calorimeter at a constant rate, secured by keeping it under a constant head, and enters continuously at a uniform temperature. Upon leaving the calorimeter, it may be weighed, and the heat added ascertained by noting its rise of temperature. In order that all the heat be given up to the water, the products of combustion must return to the temperature of the fuel and air from which they came, before leaving the calorimeter. This is ascertained by a thermometer placed in the exit gas flue. The water resulting from the burning of hydrogen is condensed and collected, should it be desired to calculate the lower heat value. (See p. 170.)

For gases, a Bunsen burner is used and a small gas meter to measure the fuel in cubic feet. For liquid fuels, a special regenerative burner is used which, together with the lamp containing the fuel, is attached to one arm of a beam balance so that the weight of fuel burned may be measured at any time during the test.

There is no water equivalent of the calorimeter to be considered since, during operation, all parts are at constant temperatures.

Radiation is provided against by air jacketing and polished surfaces, and is negligible.

Sampling. If the sample to be tested represents a gas used in a test of several hours' duration, it is best to take a continuous sample covering the whole time of the test as described on page 193, for exhaust gas sampling. This involves rather a large gas container, however, and it is more convenient to draw the gas from the main directly into the calorimeter. For this purpose the main should be tapped at a point as near as possible to the place where the gas is used. Then, if a number of heat value determinations are made covering equal intervals of time and separated by equal intervals, their average will be the average heat value of the total gas delivered in the main.

(a) Determination of Higher Heat Value. The rates of water and gas flow should be adjusted so that the temperature of the products of combustion upon leaving the calorimeter is that of the room. Then, if t and T are the temperatures of the water before and after heating, respectively, in degrees Fahrenheit, W, its weight in pounds, and G, the cubic feet of gas burned, we may say

$$\text{Higher heat value} = (T-t)W \div G.$$

In this, G is the number of cubic feet of gas under standard conditions (see p. 170). To convert the gas meter reading into standard cubic feet, it is necessary to use the relation that the volume of a gas varies directly as its absolute temperature and inversely as its absolute pressure. Consequently, the pressure as well as the temperature of the gas should be noted. This involves a reading of the barometer and of a manometer set in the gas main.

The heat value determined in this way is not quite the higher value, for the reason that not quite all the water of combustion is condensed. This is because the air entering combustion is not saturated with water vapor (although the fuel may be), but the products of combustion are saturated. The difference in the humidities is due to the water of combustion which is

not condensed, the result being unaccounted for latent heat. A correction may be computed as on page 382. The determinations as made without this correction are acceptable for most purposes.

(b) Determination of Lower Heat Value. This is determined correctly by subtracting from the heat added to the water the amount which came from the vapor in condensing, or its latent heat. To get this amount correctly, we should know the weight of the vapor and its latent heat per pound. The latent heat of steam is generally determined from its pressure by reference to the steam tables. In the case of steam mixed with other gases, the pressure of the steam is only partial (see p. 142) and therefore difficult to determine. Under the conditions existing during the operation of the Junker calorimeter, it is convenient and sufficiently accurate to take the latent heat of the steam as that corresponding to the *temperature* of the exhaust gases, that is, room temperature. Letting L stand for this heat and w for the weight of condensation, we have

$$\text{Lower heat value} = \frac{(T-t)W - Lw}{G},$$

the other notation being as used under (a).

An average value of L may be taken as 1060 for temperatures between 50° and 70° F.

(c) Calculation of the Heat Value from the Fuel Analysis. This may be readily and acceptably done in the case of gases when the complete analysis is known. The following rule may be used:

To calculate the higher or lower heat value of a fuel gas, multiply the higher or lower heat value of each of the constituent gases, in B.t.u. per standard cubic foot, by its volume percentage, as shown by the fuel gas analysis, and divide by 100. *Add these results together; the sum is the desired heat value.*

The heat values of various constituents of fuel gases are

given on page 171. An example of the calculation is given in columns 1, 2, 6 and 7 of the table on page 294.

Problem 39_1. Figure the higher heat value from a calorimeter test giving data as follows. Pressure of gas = 4 ins. of water. Barometer, 29.4 ins.; temperature of gas, 70° F.; 7.24 lbs. of water raised from 60.7° to 116.3°. Volume of gas burned, 0.875 cu. ft. by meter. *Ans.*, 500 B.t.u.

Problem 39_2. In the preceding problem, if there were .048 lb. of water of condensation, what would be the lower heat value? *Ans.*, 437 B.t.u.

Problem 39_3. Calculate the higher and lower heat values of the producer gas, analysis of which is given in the table on page 165. *Ans.*, 148 B.t.u.

THE PRODUCTS OF COMBUSTION

The products of combustion of fuels used commercially are called exhaust gases. By their analysis, we learn the completeness of combustion and the directions and amounts of heat losses in the operation of boilers, internal combustion engines, gas producers, and furnaces in general.

Commercial combustion, as previously defined (p. 170) involves the chemical combination of the oxygen in air with the combustibles of a fuel, resulting in CO_2 and H_2O. Air is principally oxygen and nitrogen, the other constituents being negligible in this analysis, and the proportions may be taken by volume, 79 of nitrogen to 21 of oxygen (more exactly, 79.1 : 20.9 = 3.78); and by weight 77 to 23. Almost always more than enough air is used than that necessary to satisfy the theoretical reaction between the combustibles and the oxygen. The exhaust gases will therefore contain free oxygen and nitrogen due to this extra air, in addition to CO_2 and H_2O. Sometimes the combustion is incomplete, in which case the exhaust gases may contain CO, free hydrogen, and hydrocarbons. With proper operation, however, these are each less than 1 per cent by volume of the total exhaust gases, and the hydrogen and hydrocarbons are almost always negligible. The experimental determinations

are therefore generally only for CO_2, CO, O_2, and N_2; H_2O being calculated from the fuel analysis. The analysis of the exhaust gases is found and expressed as percentages by volume.

The ratio, by volume, of the air actually supplied to the fuel to that required for "complete theoretical combustion," is called the "excess coefficient."

The Theory of Combustion is based upon the following relations and laws.

Atomic weights of the elements to be considered

$$H=1, \quad O=16, \quad N=14, \quad C=12$$

Elemental gases are supposed to have two atoms to a molecule. For that reason they are written O_2, N_2, H_2, etc. Carbon does not occur as a gas except in combination with other elements, but it will be convenient to conceive hypothetical gaseous carbon having a single atom to the molecule.

The molecular weight of a gas equals the sum of the weights of the atoms in its molecule. Thus H_2 weighs $1+1=2$, O_2 weighs $16+16=32$, CO_2 weighs $12+16+16=44$, and so forth.

Avogadro's Law. *In a given volume of any gas, there is always the same number of molecules, regardless of the nature of the gas and of the number of atoms to each molecule, provided the conditions of pressure and temperature are constant.*

From this it follows that the weight of a unit volume at a given pressure and temperature of a gas (or a mixture of gases) is proportional to its molecular weight (or average molecular weight, if a mixture). If the weight per unit volume, or density, is called d in pounds per cubic foot, and, m, the molecular weight,

$$d = \text{a constant} \times m.$$

Since the specific volume, V, in cubic feet per pound $=1/d$,

$$1/V = \text{a constant} \times m$$

and

$$Vm = \text{a constant}.$$

This last constant can be calculated from the relation PV(m) = (m)RT, for any pressure-temperature condition of a perfect gas. At 14.7 pounds per sq. in., abs., and 32° F. the constant is 359, and at 14.7 pounds and 62° F. it is 380.

The Mol. The quantity Vm is the volume in cubic feet occupied by m pounds of any perfect gas under standard conditions. The weight of this quantity is called a "mol," and can be defined as a quantity of a gas, numerically equal in pounds to its molecular weight. Thus at 14.7 pounds and 32°

1	mol of	H_2	weighs	2 lbs.	and	occupies	359 cu. ft.
1	" "	O_2	"	32 lbs.	"	"	359 cu. ft.
1	" "	CO_2	"	44 lbs.	"	"	359 cu. ft.

and so on.

From this follows the convenient relation: *The specific volume of any perfect gas under standard conditions equals* 359 *divided by its molecular weight.*

The word "mol" implies weight, but, since the volume of a mol of gas is constant, the mol may also be used as a measure of volumes of gases. Thus we have two sets of mol units, "molar weights" and "molar volumes." In this work, the word "mol," when unqualified, signifies weight; "mol-volume" will signify volume. For example, if two mols of CO are combined with one of O_2, and burned to make CO_2, the reaction could be written:

in molecules $O_2+2CO=2\ CO_2$ (= volumes, or mol-volumes)
or, in cu. ft. 359 of O_2 + 718 of CO = 718 of CO_2
or, in mols. 1 of O_2 + 2 of CO = 2 of CO_2
or, in lbs. 32 of O_2 + 56 of CO = 88 of CO_2.

The first three equations are the same in so far as proportions by volume are concerned. The fourth equation shows proportions by weight.

Air Required to Burn Carbon and Hydrogen

Assuming no excess air, the reaction for carbon is

$$C+O_2+\frac{79}{21}N_2=CO_2+\frac{79}{21}N_2$$

the coefficient of N_2 being the proportion, by volume, of N_2 to O_2 in air. In other words, every volume (or molecule) of O_2 used in combustion must be accompanied by $\frac{79}{21}$ volumes (or molecules) of N_2. This ratio is more conveniently applied by using its value 3.78.

If the quantities on the left of the above equation are considered to be mols, then the weight of the C is 12 lbs., and of the air is $32+3.78\times28=138$ lbs.

The weight of air used to burn one lb. of carbon, then, is $138\div12=11.6$ lbs.

Similarly, for hydrogen, $2\,H_2+O_2+3.78\,N_2=2\,H_2O+3.78\,N_2$.

The ratio by weight of air to hydrogen is $(32+3.78\times28)\div4=34.8$ lbs. of air to burn one lb. of hydrogen.

Water Vapor Formed in Combustion

Since one mol of H_2O weighs 18 lbs., and the hydrogen in it 2 lbs., the ratio of water vapor formed to hydrogen burned is 9, by weight. The water of combustion may be calculated by multiplying the weight of hydrogen per unit of fuel by 9, the result being in lbs. of H_2O from the combustion of a unit of fuel.

The application of these principles to combustion calculations will now be shown. Coal, oil, and gas fuels will be considered, each for the following items: (*a*) Quantity of air required per unit of fuel, (*b*) Products of combustion to be expected in weights or volumes per unit of fuel, assuming the air-fuel proportions entering combustion, (*c*) Calculation of combustion products per unit of fuel from the exhaust gas analysis.

Convenient formulas for the various quantities sought will be deduced under Test 40 (b).

Combustion of Coal

Air Required. If C and H stand respectively for the fraction of a pound of carbon and hydrogen in one pound of fuel, and if the coal contains no oxygen by which combustion may be supported, then the total amount of air needed per lb. of fuel is

$$11.6C+34.8H$$

which relation is sufficiently accurate for most purposes. Suppose, however, that the fuel contains O lbs. of oxygen per pound of coal. Accounting for this, the air required is

$$11.6C+34.8\left(H-\frac{O}{8}\right)$$

The introduction of the term $O/8$ is explained thus. The weights entering in the formation of H_2O are 2 of hydrogen to 16 of oxygen, or eight weights of oxygen are required to burn one of hydrogen. Consequently the oxygen in the coal will take care of $O/8$ weights of hydrogen; the difference between this amount and H being the quantity of hydrogen requiring oxygen from the air.

Products from Coal Combustion. In the burning of pure carbon, if complete combustion could take place with no excess of air, the reaction would be

$$79N_2+21O_2+21C=21CO_2+79N_2$$

showing that the exhaust gas would consist of 21 per cent of CO_2 and 79 per cent of N_2. Now, if more than this amount of air is used, the sum of the free oxygen and of the CO_2 in the exhaust gas, will bear the same relation by volume to the nitrogen as the oxygen did in the air, since the volume of the oxygen is not increased by combination with carbon. Consequently the sum of O_2 and CO_2 is 21 per cent, and N_2 is again 79 per cent.

Ash does not vary this relation, but the effect of hydrogen, oxygen, and nitrogen in the coal is to alter it. The last two have slight effect, but the hydrogen needs oxygen from the air to burn to water, and as the water does not appear in the exhaust gas analysis, a corresponding amount of oxygen disappears. This increases the percentage of nitrogen. The formation of CO instead of CO_2 has an opposite effect, since the oxygen entering the combination makes a volume of CO twice that of the O_2 entering combination with C. On the other hand, CO is generally low.

Anthracite and semi-anthracite coals, being low in hydrogen, will yield exhaust gas having between 79 and 80 per cent of N_2. Bituminous coals will yield between 80 and 81.5 per cent. Results outside of these figure are questionable.

It is obvious that the less air is used, the less will be the free oxygen in the exhaust gas, and the greater will be the CO_2. It will be shown (Test No. 54) that the less the air used the more economical is the combustion, provided that there is not so little air as to cause material loss from incomplete combustion. Consequently high CO_2 is a desirable indication.

Calculations from the Exhaust Gas Analysis. Assume a volume analysis as shown by second column in following table, and that each per cent is the volume of one mol.

	Mol-volume	weight in lbs. =	weight in lbs. of C =
CO_2..............	10	$10\times44=$ 440	$10\times12=120$
O_2...............	9	$9\times32=$ 288	
CO..............	1	$1\times28=$ 28	$1\times12=$ 12
N_2...............	80	$80\times28=2240$	
Total weight in lbs. =		2996	132

The weight of the products of combustion per lb. of carbon in the products is then $2996\div132=22.6$. If this is multiplied by the weight of carbon in one lb. of coal, the result is weight of products, ex-

clusive of H_2O, per lb. of coal. (This assumes that all of the carbon in the coal appears in the exhaust gas, which assumption will be modified later.)

In this manner may be figured the weight of any single product of combustion per lb. of coal.

Combustion of Oil

Air Required. As for coal, this is $11.6C+34.8H$. Since the values of C and H are nearly the same for all American fuel oils, the air required is very nearly the same. Thus, an average value of C is 0.84 lbs. and of H is 0.14 lbs. Then

$$11.6\times.84+34.8\times.14=13.6 \text{ lbs. of air per lb. of oil.}$$

Products of Combustion of Oil. Let us consider the C and H_2 to exist as gases just before combustion. The number of mols of C per pound of oil will then be $.84\div12=.07$. Similarly the number of mols of H_2 per lb. of oil will be $.14\div2=.07$. The ratio of C to H_2, by volume when gasified, is .07 to .07 or one to one. The reaction may now be written:

$$\begin{array}{ll} \text{Fuel}\ldots\ldots\ldots\ldots & C+H_2 \\ +\text{Air}\ldots\ldots\ldots\ldots & \underline{O_2+\tfrac{1}{2}O_2+1.5\times3.78N_2} \\ =\text{Products}\ldots\ldots & CO_2+H_2O+5.67N_2 \end{array}$$

In terms of mol-volumes, then, the products of combustion will be one of CO_2, one of H_2O and 5.67 of N_2. As the H_2O condenses in the analysis, the CO_2 and the N_2 only appearing, the volume of these products will be $1+5.67=6.67$ volumes. In per cent by volume, CO_2 is $1/6.67=15$ per cent; and N_2 is $5.57/6.67=85$ per cent.

If more than the theoretical amount of air required is assumed, the additional amount could be inserted on the left of the above equation. On the right, there would then be the same amount of

CO_2 and H_2O, but, in addition free O_2 and additional N_2, all of which could be reckoned in percentages as before.

Calculations from Exhaust Gas Analysis may be made by the same methods as for coal, the products of combustion in lbs. per lb. of carbon being multiplied by the fraction of a pound of C in one lb. of oil; to reduce to the basis of one lb. of oil.

Combustion of Gases

Air Required. When dealing with a mixed fuel gas, it is convenient to assume it to be broken up into its elements of carbon, hydrogen, and oxygen. The combustion reaction may then be written much more readily. A convenient method of doing this is shown in the table on page 294, the second column of which gives the fuel analysis. The percentages by volume may be called mol-volumes. c, h, and g represent the mol-volumes of carbon, hydrogen, and oxygen which could be obtained from 100 mol-volumes of the fuel, if in the form of its elemental components, of which the combustibles are:

Fuel.............................	$80.2C+92.5\ H_2$	
Oxygen required.....................	$80.2O_2+46.25\,O_2$	$=126.45O_2$
Oxygen already contained in fuel.....		$=\ 17.35O_2$
Oxygen required from air............		$109.10O_2$
Nitrogen accompanying this $O_2=3.78\times109.1$		$=412.4N_2$

The elemental combustibles in 100 volumes of this fuel are 80.2 volumes of C and 92.5 of H_2. The air required for combustion is $109.1O_2+412.4N_2=521.$ volumes air. Ratio of air to fuel is $521\div100$ or 5.21 cubic feet of air required per cubic foot of fuel.

The same result could be obtained by dividing the mol-volumes of the O_2 required from the air by 21, the number of volumes of O_2 in 100 volumes of air: Air required $=(c+.5h-g)\div21$.

Products of Combustion. Referring to the tabulated mol-volumes above, and including the 8.4 volumes of N_2 in the fuel (p. 294), the reaction would be:

$$\begin{array}{lllll}
\text{Fuel}\ldots\ldots\ldots\ldots & 80.2C & +92.5H_2 & +17.35O_2 & +8.4N_2 \\
+\text{Air}\ldots\ldots\ldots\ldots & & & 109.15O_2 & +412.4N_2 \\
\hline
=\text{Products}\ldots\ldots & 80.2CO_2 & +92.5H_2O & & +420.8N_2.
\end{array}$$

Since the H_2O disappears in the exhaust gas analysis, the total volume accounted for will be $80.2+420.8=501$. The per

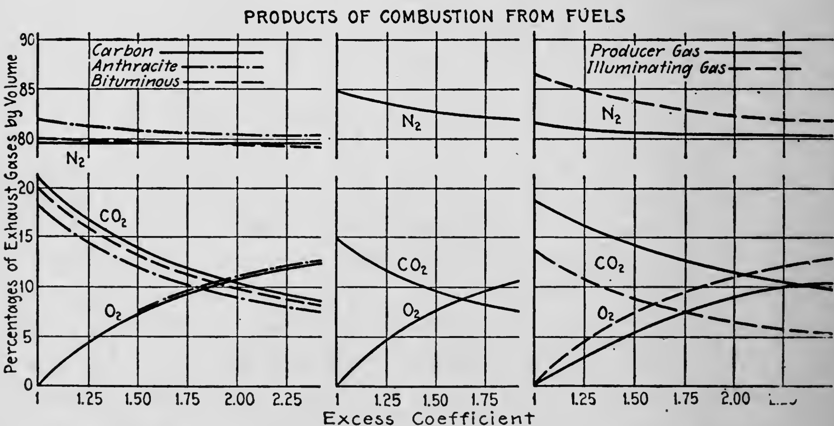


FIG. 78. FIG. 79 FIG. 80.

cent CO_2 will then be $80.2 \div 501 = 16.0$ per cent, and the per cent N_2 will be $420.8 \div 501 = 84.0$ per cent. If excess air is used, free O_2 and additional N_2 will appear on the "Products" side of the equation.

Calculations from the Exhaust Gas Analysis. This will be shown under Test 40 (b) by the deduction of general formulas.

Figures 78, 79, and 80 show the percentages of CO_2, O_2, and N_2 from coal, oil, and gas fuels, calculated by the above methods for different values of the excess coefficient.

Questions. (*a*) How much air is required to burn one lb. of the bituminous combustible given at bottom of p. 163? (*b*) How many lbs. of H_2O will be formed? (*c*) What per cent of CO_2 will be in the products of combustion if no excess air is used? (*d*) If the actual products contain 5% of CO_2, 14% of O_2, and 81% of N_2, what will be the weight of the dry products? (*e*) Answer all of the above questions except (*d*) for one cubic foot of methane (CH_4). (*f*) What is the specific volume under standard conditions of CO? (*g*) What is the density? (*h*) What is the specific volume of the producer gas on page 165? (*i*) Write the reaction, as on page 187, for the producer gas on page 165.

40. The Analysis of Exhaust Gas

Principles. The gas analysis apparatus depends upon the separate absorption of the products of combustion by certain reagents. A measured volume of the gas is brought into contact with one of these reagents which removes the CO_2, but does not take up any of the other constituents. The CO_2 is then determined by measuring the diminution in volume. In this way the volumes of CO and O_2 may also be found. The residue is assumed to be N_2. It is convenient to make the original volume 100 units, as cubic centimeters, in which case the differences are per cents.

The Orsat apparatus in its various forms is in general use for engineering analyses of exhaust gas. It consists of a measuring flask, B', called the burette (see Fig. 82), a distributing tube, T, and three so-called pipettes, DD', OO', and MM', containing the reagents. The gas sample is displaced from the burette into each pipette in turn by filling the burette with water from the bottle B. The right leg of the pipette is completely filled with reagent just before the gas enters; the gas displaces it into the left leg. Rubber bags, R, seal the left legs of the pipettes

so that the reagents will not deteriorate through contact with the atmosphere.

If desired, the apparatus may be provided with a fourth pipette for determining hydrogen.

There are many other types of exhaust gas analysis apparatus,

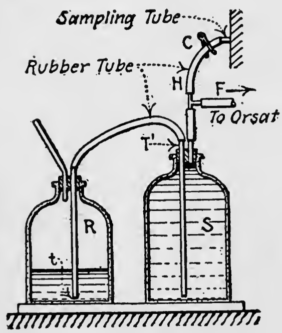


Fig. 81.—Gas Sampling Bottle.

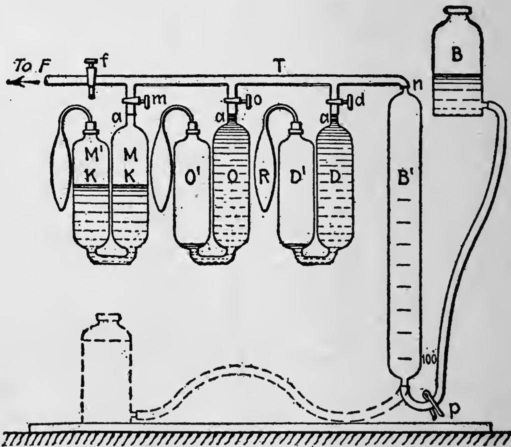


Fig. 82.—Orsat Apparatus.

the most important of which are recording instruments giving the percentage of CO_2 in boiler flue gas continuously.

CO_2 Recorders.* These may be divided into two classes, continuous and intermittent. Of the former, a representative one is the Uehling. Its principle is shown diagrammatically by Fig. 83. A steam aspirator draws the gas through the two orifices, *A* and *B*, between which is a special, dry absorbent of CO_2. If no CO_2 is present, there will be a definite pressure, below atmosphere, between the two orifices. If CO_2 is present

* For methods of installing and adjusting various types of CO_2 recorders, see Bulletin 91, Bureau of Mines, on *Instruments for Recording* CO_2 *in Flue Gases,* by Barkley and Flagg.

it is absorbed between the orifices, reducing the volume of gas in this space, and thereby decreasing its pressure. Pressure therefore becomes a measure of CO_2 removed, and is recorded by a low pressure recording gage calibrated in percentages of CO_2.

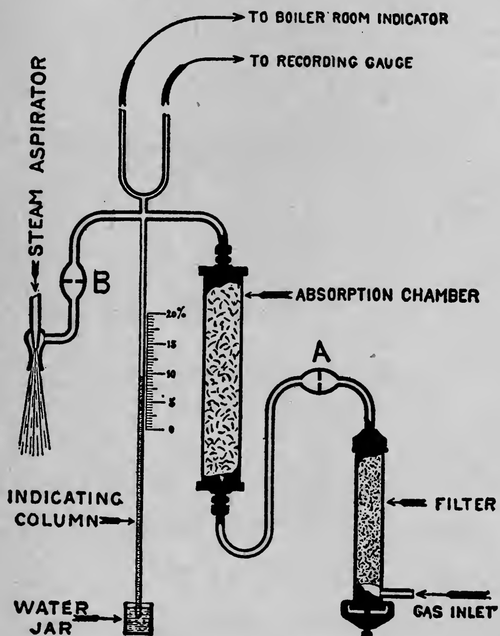


FIG. 83.—Diagram of Uehling CO_2 Analyzer.

The instrument is provided with a regulator for automatically maintaining a uniform flow.

The Hays Automatic CO_2 Recorder is of the intermittent type. Referring to Fig. 84, gas is drawn from the furnace and through the machine by means of a water aspirator. The same water then flows into the standpipe to furnish the motive power, it being intermittently removed by the syphon. As this water rises, a sample of gas is trapped off, placed at atmospheric pressure, and an

accurately measured portion collected in the measuring burette. This is pushed over into the absorption chamber where the freshly exposed surfaces of steel wool, wet with caustic potash solution, break up the gas train and absorb the CO_2 content. A quantity of caustic potash solution is thus displaced into the caustic jar, and a like quantity of water forced out of the rubber bag into the compression cylinder.

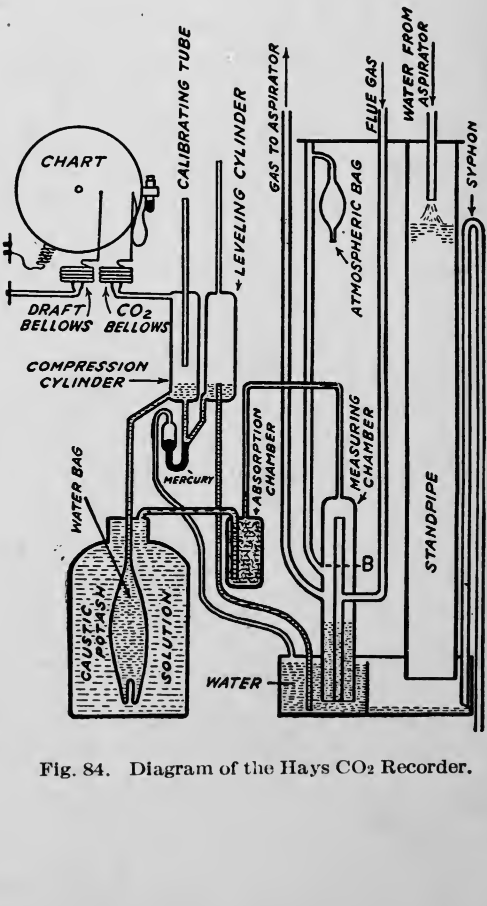


Fig. 84. Diagram of the Hays CO_2 Recorder.

The height to which the water rises in the compression cylinder is a measure of the amount of CO_2 absorbed. The calibrating tube is so placed in the standard machine that it becomes sealed if exactly 20 per cent of the sample is CO_2. If the CO_2 content is less than 20 per cent there is a compression of air in the CO_2 bellows system and a corresponding movement of the pen on the chart. (The draft pen is operated independently by the draft bellows.)

Gas is drawn from the furnace continuously, being bypassed through a tattler jar, while the recorder is analyzing a sample. Thus the machine always receives a fresh sample.

The mercury valve causes the water in the compression cylinder to be releveled after every analysis.

Reagents for Orsat Apparatus. For CO_2, a 1 to 2, by weight, solution of potassium hydroxide (KOH) or sodium hydroxide (NaOH) in water. The reagent will absorb 40 times its own volume of CO_2.

For O_2, the reagent is a mixture of water, potassium hydroxide, and pyrogallic acid.

This is prepared by dissolving one part, by weight, of KOH or NaOH in two of water, making solution "A." Another solution "B" is prepared by dissolving 1 part of pyrogallic acid, by weight, to three of water. Equal volumes of solutions *A* and *B* should be mixed to make the reagent for O_2. The absorption capacity is twice its own volume.

For CO, the reagent is a 3 to 20 solution of cuprous chloride (CuCl) in water. To this should be added just enough strong ammonia to cause a blue color. The absorptive capacity is four times its own volume.

Solutions *A* and *B* may be mixed in quantity, *A* being useful either for CO_2 as mixed or for preparing the O_2 reagent. The CuCl may also be kept in solution without the addition of the ammonia.

Sampling. For use in the apparatus, a small amount of the total gas is led from the main body through a sampling tube. The proper construction of this tube depends upon whether the gas is flowing in a large flue, as in boiler work, or in a com-

paratively small pipe, as in gas engine work. For the latter it is sufficient to tap into the exhaust pipe a pipe of about $\frac{1}{4}$-inch diameter. In a boiler flue, however, the composition of the gas may vary through any section of the current, so means should be provided for drawing gas from various parts of the section. One method of doing this is to run the sampling pipe diagonally across the flue, the end of the pipe being sealed and holes being drilled across the length in the flue. The holes should increase uniformly in size from the point of entrance of the sampling pipe to its end, as otherwise more gas will be drawn from the holes in the near end of the pipe. The pipe should enter as nearly as possible to the furnace, but beyond the combustion chamber or last pass in which combustion is taking place. This should be done because the farther the gas gets from the furnace the greater is its opportunity to become diluted with air entering through leaks in the brick work or setting.

The sample may be collected with an apparatus sucn as shown by Fig. 81, consisting of two 3- and 5-gallon flasks connected as shown. The flask S is used to receive the sample. To start with, it is filled with water. During collection, the water is syphoned into flask R through T' and the gas flows through H into the space evacuated, branch F being closed.

All the air in the tube H should be displaced before taking the sample. This may be done either by raising the flask R above the level of the sampling tube so that water will syphon into the the tube H, or by drawing a small charge of gas into flask S in the regular manner and then discharging it through a three-way cock at the end of the branch F. The latter procedure leaves flue gas in the tube H very slightly diluted with air.

The sample should be collected at a uniform rate, and, if more than one sample of the same gas or for the same test is to be analyzed, time periods during which collections are made should be equal in order to get a correct average result. The rate may be kept practically uniform by placing flask R several feet below S and

by choking the end of the syphon at *t* with a sliver of wood. The change in the rate due to the decreasing distance between the water levels in the flasks may then be inconsiderable.

For boiler and gas engine tests, it is well to collect the sample during half hour or hour periods, and during the collection of each sample to analyze the preceding one. This has the advantage of showing the uniformity of combustion as the test proceeds. It is also well to have a duplicate sampling outfit, collecting gas at a very slow rate so that the sample will cover the whole test. This may be analyzed at the convenience of the experimeter; the results should check the average of the separate analyses.

(a) Determination of CO_2, O_2, CO and N_2. The Orsat apparatus is first connected with a rubber tube to the branch *F* leading from the sample bottle (see Figs. 81 and 82). The reagents in the pipettes are brought to the levels *a*, by manipulation of the bottle *B*, the three-way cock *f* being closed. This being done, and the cocks, *d*, *o*, and *m*, closed, the water level in the burette is raised to the point *n*, thus expelling the air previously contained through the three-way cock *f* which is open to the atmosphere for that purpose. The pinch-cock *C* on the sampling bottle flue connection is now closed, and the flask *R* raised so as to put the sample under pressure. The cocks between the sample bottle and the burette are then opened, and the bottle *B* lowered so that part of the sample will flow into the burette. The gas now in the burette is diluted with air or other gas which was in the distributing tube and in the rubber tube leading to the sampling bottle. To get a purer charge, this first one is discarded through the three-way cock, *f*, and another one brought into the burette, in the same way as the first, which may be accepted for analysis. This charge should be somewhat greater than 100 cc.

Having secured this sample by closing the three-way cock, the bottle *B* is raised a trifle until the water level in the burette reaches the lowest graduation which marks 100 cc. of gas. As

the sample has been compressed by this rise in water level, the pressure should be made equal to atmospheric by opening the three-way cock for an instant, the pinch-cock *p* being closed so as to maintain the water level at the lowest graduation.

The sample should now be worked back and forth in the KOH solution to remove the CO_2. After this has been done for a few minutes, it should be returned to the burette, care being taken that the reagent is at its original level, *a*; the cock *d* should be closed; and a volume measurement of the gas made. This must be done with the sample under the same condition of pressure as when originally measured, that is, atmospheric. For this reason, the water levels in the burette and in the bottle *B* should be coincident when the reading is made. Before making the reading it is well to allow the walls of the burette to drain for a minute. After the reading the gas is returned to the KOH solution and worked a few more times, then measured again. If the volume is the same the result is satisfactory, if not, the process should be repeated until all the CO_2 has been removed as indicated by constant volume. The diminution of volume is the per cent CO_2.

The O_2 and CO are removed and measured in the same way and in the order named. The difference between the sum of the per cents of CO_2, O_2, and CO, and 100 may be taken as the percentage of N_2.

Precautions in Operating. Before using the apparatus, all connecting tubes and cocks should be tested for leaks by putting air in the system under pressure or partial vacuum with the bottle *B*, and noting whether the volume remains constant as shown by the level in the burette.

Analyses should be made at a uniform temperature of not less than 60° F. During operation, the apparatus should not be exposed to a changing temperature, or placed where drafts or sun may strike it. The corresponding changes in the volume of the sample would give false results.

Water absorbs all of the gases to some extent. To avoid the resulting errors, it is well to use water that has been previously saturated with exhaust gas, so that it will not take up any more. To do this the exhaust gas is caused to bubble through the water used in the sampling flask and in the burette.

If a little coloring matter, such as red ink, is added to the water used in the burette, it will aid in the reading of the graduations.

It is well to keep a record of the volumes absorbed by each reagent so that it will be known when their absorptive capacity has been reached. The volume of reagent in a pipette is about 150 cc.

(b) Calculations from Exhaust Gas Analyses. The principles upon which these depend are stated and illustrated on pages 180 to 183. They should be thoroughly understood.

The results for coal combustion are generally based upon 1 lb. of dry coal, while those for fuel gas are based on one standard cubic foot of the fuel. Therefore two different sets of formulas will be deduced for the two cases.

It is convenient to apply the following rule for a number of purposes. *The specific volume of a gas, in standard cubic feet per pound (that is, at 14.7 lbs. per square in. and 32 deg. F.) equals 359 divided by the molecular weight of the gas (see page 181). Conversely, its density equals its molecular weight divided by 359.*

The following notation will be used for the two cases in common:

D, M, O, N, H = percentages of carbon dioxide, monoxide, oxygen, nitrogen, and hydrogen, by volume, in the exhaust gas, respectively.

V_d = volume of dry exhaust gas in cubic feet per pound of coal or per cubic foot of fuel gas.

W_a = weight of air supplied in pounds per pound of coal, or per cubic foot of fuel gas.

W_d = weight of dry exhaust gas in pounds per pound of coal, or per cubic foot of fuel gas

W_v = weight of H_2O in the exhaust gas, per pound of coal, or per cubic foot of fuel gas.

X = excess coefficient; that is, the ratio of the amount of air supplied to that required for complete theoretical combustion.

It should be borne in mind that the weights here tabulated are quantities resulting from the *combustion of* 1 *lb. of dry coal*, or from 1 *standard cubic foot of fuel gas* (see page 170).

Formulas for Coal Combustion. In 100 mol-volumes of the exhaust gas there are $D+M$ mol-volumes of carbon, since each per cent of CO_2 and CO contains 1 mol-volume of carbon. Therefore, the ratio $100 \div (D+M)$ is the number of cubic feet of dry exhaust gas per cubic foot of gaseous carbon contained by it. This is for *dry* exhaust gas (that is, the volume of the water actually contained is omitted), because the analysis does not take account of the H_2O. If multiplied by the volume of 1 lb. of gaseous carbon ($=359 \div 12 = 29.8$), the result is the number of cubic feet of dry exhaust gas per pound of carbon in it, or $2980 \div (D+M)$. Multiplying this by the weight of carbon that is gasified from 1 lb. of dry coal, we have the volume of dry exhaust gas per pound of dry coal, or

$$V_d = 2980 \frac{C_g}{D+M},$$

in which C_g is the weight of carbon gasified. This quantity is all of the carbon in 1 lb. of dry coal, C_t, except that lost through the grate and removed with the ash, C_a; or

$$C_g = C_t - C_a$$

assuming carbon deposited as soot to be negligible. C_t may be estimated from the proximate analysis and C_a measured from the

total ash (Test 54(c)). If C_t is found from the analysis in terms of percentage of the coal as analyzed, that is, wet coal, it should be divided by 100 minus the percentage of moisture, to reduce to weight per pound of dry coal.

To find the weight of air supplied per pound of dry coal, we may assume that all of the nitrogen in the exhaust gas comes from the air, a very nearly correct assumption, since coal contains but little nitrogen. Now, there are 28 N pounds of N_2 and 12 $(D+M)$ pounds of carbon in 100 mols of the exhaust gas. Therefore, the weight of N_2 supplied by the air to each pound of carbon gasified is $\frac{28N}{12(D+M)}$ pounds. Dividing this by the proportion of N_2 in air, 0.77 by weight, gives the pounds of air supplied to a pound of the carbon in the exhaust gas. Multiplying by the weight of carbon gasified from 1 lb. of dry coal gives the weight of air supplied per pound of dry coal, or

$$W_a = \frac{28NC_g}{12\times .77\times(D+M)}$$

$$= 3.04\,\frac{NC_g}{D+M}$$

The 1922 Boiler Test Code of the A. S. M. E. gives another relation for W_a, namely

$$W_a = W_d + 9H_t - C_g$$

which, interpreted, states that the weight of air actually used equals the weight of dry products plus water of combustion minus the carbon gasified.

The weight of the dry products are as calculated on page 185, which can be generalized thus:

$$W_d = \frac{44D+32O+28N+28M}{12(D+M)}$$

$$= \{11D+8O+7(N+M)\}\div 3(D+M)$$

Water vapor in the exhaust gas comes from three sources, namely from combustion of hydrogen, from moisture in the coal, and from the moisture in the air supplied. The first two only need be considered.

If H_t is the weight of hydrogen in 1 lb. of dry coal, the resulting water vapor, if all the hydrogen is burned, will be $9H_t$, since the ratio of weights of H_2O to the H_2 in it is $(2+16)\div 2=9$. If m is the weight of moisture in the coal per pound of dry coal, then the total weight of H_2O in the exhaust gas, per pound of dry coal is

$$W_v=9H_t+m$$

The assumption that all the H_2 is burned is sufficiently accurate under usual operating conditions.

If H_t and m are found from the proximate analysis (see Test 37, (*a*) and (*b*)) as percentages of the coal as analyzed, they should be divided by 100 minus the percentage of moisture, to reduce to pounds per pound of dry coal.

The weight of carbon incompletely burned, per pound of dry coal, may be calculated from the CO appearing in the exhaust under the assumption that there is no incomplete combustion through unburned hydrocarbons. The method is the same as for the determination of the weight of air. Let W_t be the desired weight Then

$$W_t=\frac{12MC_g}{12(D+M)}$$

$$=\frac{MC_g}{D+M}$$

To find the excess coefficient, it is necessary to divide the value of W_a, as deduced above, by the expression for the air required as given on page 184, then, neglecting O_2 in the coal,

$$X=W_a\div(11.6C_t+34.8H_t).$$

Another expression for X, based upon the air needed for combustible burned (instead of coal fired) is $X=\frac{N}{3.78(O-.5M)}$, the derivation of which will easily be seen.

The formulas above also apply to the **combustion** of **oils** if C_t is substituted for C_g, and if m is left out of the expression for W_v.

Formulas for Gas Combustion. These are deduced upon the assumption that all the carbon in the fuel appears in the exhaust gas analysis, and that none other does. Contrary to this, in the case of internal combustion engines, is the fact that some carbon is left as a deposit in the cylinder, but it is so small compared with the total carbon used as to be negligible. Also, if the lubricating oil burns, it will appear as CO_2 in the exhaust, so care should be taken to avoid this condition. Hydrocarbons in the exhaust, too, will prevent the correct application of the formulas. If the CO in the exhaust is less than 1 per cent, it is very unlikely that there is any hydrogen or hydrocarbon, because CO is the least readily burned of the fuel gas constituents.

The following relation will be used to get most of the formulas.

Weight or volume of the substance sought, per cubic foot of fuel gas = weight or volume of that substance contained in 1 cu. ft. of exhaust gas × volume of exhaust gas per cubic foot of fuel gas.

If an expression for the last named quantity, which is V_d, be deduced, it will remain only to find the required quantities per cubic foot of exhaust gas.

The following additional notation will be used.

c, h, g = the number of mol-volumes in 100 mol-volumes of the fuel gas, of carbon, hydrogen, and oxygen, respectively; counted as in columns 3, 4 and 5 of the table on page 294.

R = Ratio, by volume, of air supplied to gas, that is, the number of cubic feet of air per cu. ft. fuel.

To find V_d. Since there are $D+M$ mol-volumes of carbon in 100 of the exhaust gas, and c mol-volumes in 100 of the fuel gas,

Cubic feet of exhaust gas per cubic foot of gaseous C in it $= \frac{100}{D+M}$

Cubic feet of fuel gas per cubic foot of gaseous C in it $= \frac{100}{c}$

By assumption, the gaseous carbon is the same in both gases. Hence, by division,

Cubic feet of exhaust gas per cubic foot of fuel gas =

$$V_d = \frac{c}{D+M}$$

The volumes of exhaust gas above are for dry gas.

The volume of air supplied per cubic foot of fuel gas is found as follows. The oxygen appearing in 100 mol-volumes of the exhaust gas is $D+O+.5M$ mol-volumes. Dividing this by .21, the proportion of oxygen in air by volume, gives the corresponding amount of air; and the cubic feet of air which supplied this oxygen, per cubic foot of exhaust gas, is $\frac{D+O+.5M}{100\times.21}$. Multiplying by V_d, gives the cubic feet of air per cubic foot of fuel, that came in with the oxygen that *appears* in the exhaust gas. But some oxygen has disappeared in the form of water. As there are h mol-volumes of hydrogen in 100 of the fuel, and as each one combines with half its volume of O_2, $\frac{0.5h}{100}$ is the cubic feet of oxygen disappearing as water, per cubic foot of fuel. But some of the oxygen comes from the fuel itself, and we wish only that which comes from the air. Subtracting the mol-volumes of oxygen in the fuel gives $(.5h-g)\div 100$. Dividing by 0.21, we get

the corresponding air. Then the cubic feet of air supplied per cubic foot of fuel gas is,

$$R=\frac{V_d}{21}(D+O+.5M)+\frac{.5h-g}{21}$$

If there is H per cent of hydrogen in the exhaust, too much oxygen has been counted. To allow for this, if the free hydrogen is known, subtract $0.5H$ from the quantity in the parenthesis.

To find W_d. One hundred mols of the dry exhaust gas weigh $44D+32O+28M+28N$ lbs. The volume of 100 mols is 100×359 cu. ft. The weight per cubic foot of the dry exhaust gas is therefore

$$\frac{44D+32O+28(M+N)}{359\times100}$$

Multiplying this by V_d, and simplifying, we have, very nearly,

$$W_d=\frac{V_d}{9000}\{11D+8O+7(M+N)\}.$$

To find W_v. Besides the water of combustion, there is some water vapor in the exhaust due to moisture in the air and fuel gas. These will be disregarded, as they have but a slight effect upon the heat analysis, owing to the fact that latent heat is not lost to the H_2O of humidity.

Since there are $\frac{h}{100}$ cu. ft. of hydrogen per cubic foot of fuel, and since the weight of the resulting H_2O is 9 times the weight of the H_2, we have,

$$W_v=9\times\frac{h}{100}\times\frac{2}{359}$$
$$=.0005h.$$

If hydrogen in the exhaust has been analyzed,

$$W_v=.0005(h-HV_d)$$

since free hydrogen would mean a corresponding decrease of H_2O.

The cubic feet of unburned CO per cubic foot of fuel equals

$$V_i = MV_d \div 100.$$

If hydrogen has been found, it may be expressed similarly.

To find the excess coefficient, the value of R, as deduced above, should be divided by the volume of air required per cubic foot of fuel, as determined on page 187, being equal to $(c+.5h-g)\div 21$. Consequently,

$$X = \frac{21R}{c+.5h-g}.$$

Problem 40₁. Draw a set of curves like Fig. 78 for the semi-anthracite coal whose ultimate analysis is C = 0.80, H = 0.04, O = 0.03, Ash = 0.10.

Problem 40₂. How many pounds of air are required to burn 1 lb. of the coal in the last problem? In this calculation, what per cent of error is involved if the oxygen contained in the coal is ignored? *Ans.*, 10.5 lbs.

Problem 40₃. How many cubic feet of air are required to burn 1 cu. ft. of the producer gas, analysis of which is given on page 165? What is the weight of this air? *Ans.*, 1.07 cu. ft.

Problem 40₄. Figure the density of the producer gas of the last problem by the molecular weight method, page 197. *Ans.*, .0703 lb.

Problem 40₅. An exhaust gas analysis (coal) gives CO_2, 6 per cent; O_2, 13 per cent; CO, 1 per cent; N_2, 80 per cent. What are the weight and volume of total oxygen in it per pound of carbon gasified? Why would not a calculation for the air supplied, based on this result, give the same value as one based on the N_2 in the exhaust gas? *Ans.*, 7.43 lbs.; 83.1 cu. ft.

Problem 40₆. An exhaust gas analysis from the illuminating gas in the table on page 165 gives CO_2, 5 per cent; O_2, 10 per cent; CO, 1 per cent; N_2, 84 per cent. What are the values of R, W_d, W_v, V_i, and X?

PART THREE

THE TESTING OF POWER PLANT UNITS

41. The Determination of Cylinder Clearance

Principles. Linear clearance of a piston engine may be defined as the least distance between the piston and cylinder head in a direction parallel to the center line of the cylinder, when the engine is on dead center. It may have different values at the head end and crank end of the stroke.

Volumetric clearance is the cylinder volume between the piston and nearer cylinder end, which may be occupied by the working medium when the engine is on dead center. It may have different values on the two ends. The volumetric clearance is not only that in the bore of the cylinder, but includes the volume of the ports up to the valve face and the volume of any fittings, such as indicator piping, which may be filled with the working medium in the usual operation of the engine. The space in cylinder drain pipes open to the cylinder is sometimes considered as clearance volume; in some cases, however, this space, in usual operation, is filled with water of condensation so that it is not part of the clearance.

Volumetric clearance is expressed as a part or per cent of the piston displacement.

"Piston displacement" is the volume swept through by

one stroke of the piston, that is, the area of the piston times the length of stroke.

A knowledge of the clearance of engines is necessary to an analysis of their performance and losses, especially in connection with indicator diagrams. See Tests 44 (*a*) and (*b*).

(a) Linear and Volumetric Clearance by Linear Measurements. The most accurate way to determine linear clearance, if the engine is not too large, is to loosen the connecting rod when the piston is at the end of its stroke, and then note how far the piston rod moves when it is pushed from the dead center position to contact with the cylinder cover. If the engine is too large for this the cylinder cover may be removed, and the measurement made by compressing between it and the piston, a small quantity of putty. The putty should be stuck to the cover at the part where the distance between it and the piston, or the piston rod extension, appears to be least. The piston should first be oiled and then dusted with graphite, so that the putty will not stick to it. The putty is then compressed by bolting the cover in place. The least thickness of the putty is the linear clearance. It may be measured by the use of a piece of fine, stiff wire.

The volumetric clearance may be obtained by mensuration from the drawings of the engine. This gives only an approximate result since the actual castings may differ materially from their drawings, and since the position of the piston in the cylinder may change with wear of the connecting rod bearings. If mensuration is used, it is best to make drawings of the engine clearances from the engine as it stands, as nearly as possible to scale.

(b) Volumetric Clearance by Water Measurement. The engine should be put on the desired dead center by putting the crank and connecting rod in line by eye. The steam valve or valves should be made tight either by smearing its face with heavy cylinder oil or, with slide valves, by squeezing a piece

of rubber gasket between the valve face and seat. The piston and cylinder bore should be well oiled to prevent leakage. If the cylinder has holes tapped on the top for indicator piping, they may be used for the introduction of water by which the clearance volume may be completely filled. By taking the weight of the water before and after filling, the amount necessary to fill the clearance space is determined. Dividing this by the density of the water gives the clearance volume. The density may be taken as 0.0361 lb. per cubic inch at 60° F. If the temperature is materially different from this, it should be noted and the corresponding density used.

Should there be a slight amount of leakage which cannot be prevented, it may be corrected for by determining the " leakage rate." This is done by measuring the amount of water necessary to keep the clearance space full for a period of one minute. Now, this is probably the maximum rate of leakage, since, during the filling, the leakage probably varied with the head of water in the cylinder. The average leakage may be taken as one-half that shown by the test. The period of time necessary for the filling should therefore be multiplied by one-half the maximum leakage rate to get the total leakage that occurred during filling. This weight subtracted from the weight used in the filling gives the weight corresponding to the actual clearance volume.

It should be noted that there are several assumptions in the foregoing which make the method approximate; it should not be used when the leakage is very large.

If the cylinder to be tested contains pockets in which air may be trapped during the filling, as is likely to be the case with internal combustion engines, these pockets should be blanked off and measured separately if possible. If this is not possible, the water method should not be used.

For accurate results, several determinations should be made and none accepted unless they show fair agreement.

To express the clearance volume in percentage, the piston displacement must be known. The bore may be measured with inside calipers, and the stroke by measuring the distance between dead center positions of the cross-head, a mark being made on the cross-head guides at each end to indicate these positions.

(c) Volumetric Clearance from the Indicator Diagram. Professor Paul Clayton proposed a method based on the fact that gases and vapors expanding or being compressed in closed cylinders follow the law that

$$\text{Absolute pressure} \times \text{volume}^n = \text{a constant.}$$

This law is represented by a logarithmic curve. Consequently if the amounts of the pressures and volumes obeying it are plotted on logarithmic coordinating paper (the coordinates being proportional to the logarithms of their markings as with the scale of a slide-rule), the result is a straight line. In an indicator diagram, the volumes represented are the total volumes of the working medium behind the piston, which include that of the clearance space. In order to transfer the indicator diagram to logarithmic coordinates, then, it is necessary to know the clearance volume so that the axis of abscissas may be located. If the indicator diagram is located on logarithmic coordinates by assuming a value of the clearance volume, the resulting expansion and compression curves will be concave to the origin, if the assumed value is too small, and convex if too large. This provides a method of determining the clearance volume, since it is only necessary to plot the expansion or compression curve from the indicator diagram on the logarithmic scale using a number of assumed values of the clearance volume until that one is found which produces a straight line. A convenient procedure is as follows.

Divide the indicator diagram with equidistant vertical lines into ten spaces, and mark the points of intersection of the verticals

with the expansion curve 1, 2, 3, etc. The coordinates of these points will be used to locate the logarithmic diagram. The absolute pressures corresponding to these points should be measured from the zero pressure line which is laid off 14.7 lbs. below the atmosphere line. After choosing a suitable scale with which to represent these pressures on the logarithmic chart, their positions are indicated by placing small marks against the vertical axis of logarithmic coordinates. Now, let a value of the clearance volume, x, be assumed in per cent. Then the point, 1, corresponds to a volume of 10 plus x per cent; point 2, to a volume of 20 plus x per cent, and so on. Using these volumes expressed as percentages, the points are readily located on the logarithmic chart. If the resulting curve is concave to the origin, a larger value of the clearance is assumed, and so on until that value which produces a straight line is found.

It will be noted that there are limiting values of the assumed clearance between which the straight line condition seems to be satisfied. On this account the method is recommended only where the clearance is large, as in internal combustion engines; the percentage of error then being not excessive.

Problem 41_1. Following are the data from a test for volumetric clearance by the water method. Weight of bucket and water before filling, 12 lbs. $14\frac{3}{4}$ ozs. Weight after filling, 10 lbs. $2\frac{1}{2}$ ozs. Time of filling, 95 seconds. To keep clearance space full for 60 seconds, 8 ozs. What is the average leakage rate in pounds per minute? What is the clearance in cubic inches allowing for leakage? What is the per cent of clearance if the engine is 10 ins.$\times$16 ins. (bore$\times$stroke)? *Ans.*, 5.22%.

Problem 41_2. In the preceding problem room temperature 60° F. of the water is assumed. How much percentage of error would be involved if the actual temperature were 40° F.? If 80°? *Ans.*, 0.1%, 0.3%.

42. Valve Setting of a Simple Slide Valve Engine

Principles. It is assumed that the student is acquainted with the principles and operation of the simple slide valve and linkage and with the various quantities pertaining, such as steam lap, exhaust lap, angular advance, etc., which subjects are amply covered by various works on the steam engine.

With a finished steam engine, in order to set the slide valve so as to better the distribution of steam, there are only two parts that may be adjusted without altering the design of the parts, namely, the valve stem, which may be changed in length, and the eccentric, which may be changed in its position on the shaft relative to the crank. The former adjustment affects the laps and lead of the valve; the latter, the angular advance. A study of the valve motion will show the effects of such adjustments upon the valve and upon the steam distribution as follows.

	Effect of Increasing.			
	Angular Advance.		Stem Length.	
On	Head End.	Crank End.	Head End.	Crank End.
Outside lap	Same	Same	Increases	Decreases
Inside lap	Same	Same	Decreases	Increases
Lead	Increases	Increases	Decreases	Increases
Admission	Earlier	Earlier	Later	Earlier
Cut-off	Earlier	Earlier	Earlier	Later
Release	Earlier	Earlier	Earlier	Later
Compression	Earlier	Earlier	Later	Earlier

The student should confirm this table by drawing the linkage as in Fig. 85 to represent the positions at the various events of the stroke, taking them first for one end of the cylinder and then the other, and observing from the drawings the effects of

the adjustments as tabulated. In Fig. 85 the engine is shown at crank end release. The valve stem is broken so as to show the valve on top of the cylinder, for convenience, instead of at the side.

It should be noted that the laps may be measured when the eccentric is vertical, as this position of the eccentric puts the valve practically in its mid-position.

(a) **Measurement of Lead.** This is made by measuring the amount of port opening, the valve chest cover being removed, when the engine is on dead center, that is, when the crank and connecting rod are in line. The engine should not be put on dead center by noting the extreme position of the cross-head

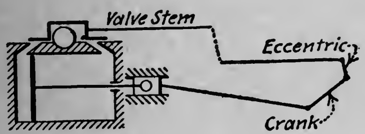


FIG. 85.

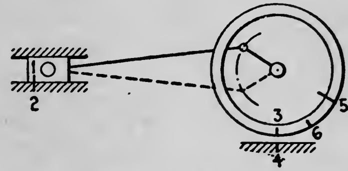


FIG. 86.

because, at the end of the stroke, there is no appreciable movement of the cross-head when the crank moves through several degrees. The eccentric, however, is then in a position where a few degrees of error will cause a considerable error in the position of the valve. Consequently, the dead center must be accurately located. The following method may be used. (See Fig. 86.)

The crank is turned until it is about 30 degrees from the dead center position sought. A mark, 1, is then made on the cross-head against a mark, 2, on the guides. Another mark, 3, is made on the flywheel against a mark, 4, on some stationary object. Now the engine is turned through its dead center position until the cross-head mark 1 again comes into the coincidence with 2. The crank and connecting rod will then occupy the positions shown by the dotted lines and the mark 3 will be at

5. A new mark is now made on the flywheel opposite 4, and the distance between this and 5 on the flywheel rim is bisected by the line 6. If the flywheel is then turned until 6 is opposite 4, the engine will be on dead center. Care should be taken that the motion of the linkage, when advancing to the positions for marking, should always be in the direction of the dead center sought so as to avoid error through lost motion of the parts. If there is no place conveniently near on which to put the mark 4, a pair of trammels should be used.

(b) Setting the Valve for Equal Leads. From the table given under "principles," it is seen a change in the angular advance will make the leads on both ends larger or both smaller, while a change of the stem length will make the lead on one end larger and on the other smaller. It will therefore be convenient to change the stem length to *equalize* the leads, and then to adjust the angular advance until they become the desired *amount*. The procedure is as follows.

The eccentric is first put in its approximate position, that is, about 135° ahead of the crank. The leads are then measured according to (*a*). If the lead on the head end is greater than on the crank end, the stem length must be increased to equalize them, and vice versa (see table). If the leads are then both too small, the eccentric is shifted on the shaft further away from the crank until the desired leads are obtained, and vice versa if the leads are too large.

The proper amount of lead depends upon the travel of the valve, the engine speed, and the amount of compression. More lead is required for high speed engines, and less when the compression is high. For low speed engines, leads up to $\frac{1}{16}$ in. are usual, and for high speeds, twice that amount.

It is advisable, after setting for equal leads, to indicate the engine as a check upon the setting. The lead should be such as to give a vertical line at the admission for both head and crank ends.

(c) Setting for Equal Cut-offs by the Bilgram Diagram. The typical Bilgram diagram is shown by Fig. 87, the positions of the crank corresponding to the various events of the stroke being shown (for the head end only) by the radial lines. The events of the stroke for the crank end may be found by reference to the crank end lap circles with similar construction. For convenience the lower half of the diagram may be revolved through 180° so as to be superposed on the upper half. For other details and for proof, see Halsey's *Valve Gears.*

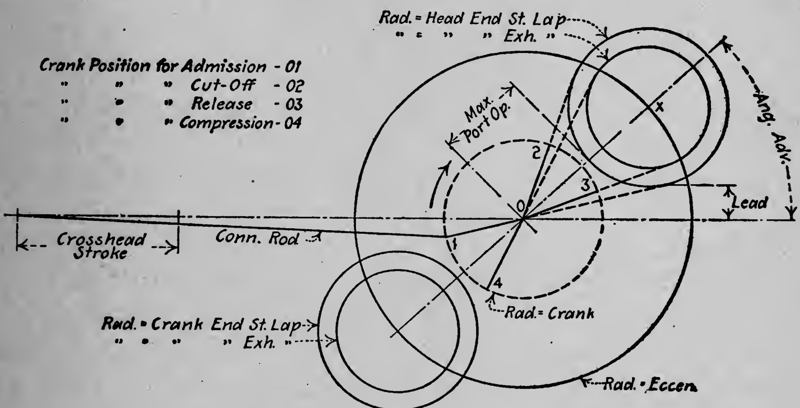


Fig. 87.—Bilgram Valve Diagram.

To lay out the diagram for the purpose of valve setting, we must assume a cut-off, and measure the connecting rod and crank lengths and eccentricity from the engine. If the cut-off is C inches from the end of the stroke, the corresponding crank positions, 01 and 02, may be found by drawing the engine linkage to scale as in Fig. 88. The diagram may be constructed on these lines, starting by putting in the eccentric circle as shown.

The values of the steam laps will depend upon the setting of the valve, but we may locate the lap circles by determining the sum of the steam laps. This equals the length of the valve

in the direction of its motion minus the distance between the outside edges of the ports, in the case of the D-valve taking steam outside. The steam lap circles may be constructed by first locating a point x on the eccentric circle such that the sum of its perpendicular distances from the lines 01 and 02 equals the sum of the steam laps. The lap circles may now be drawn about x as a center and tangent to the lines 01 and 02. The leads and maximum port openings are then determined as shown. If the leads are too large, a later cut-off must be assumed and the construction repeated.

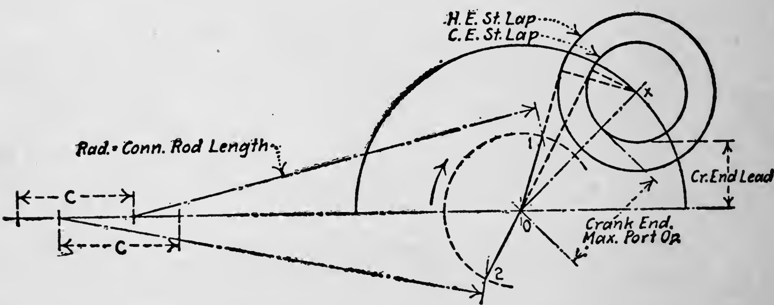


Fig. 88.—Setting for Equal Cut-offs.

Using the data for the lead and port opening at one end (preferably the crank end, since there they are larger) the valve may now be set for the assumed cut-offs. To do this, the crank is revolved until the valve has moved as far as it will toward the head end of the cylinder, thus giving the maximum port opening on the crank end and making the eccentricity parallel to the engine center line. The stem length is then adjusted until the crank end port is open an amount equal to that indicated by the diagram. Next the engine is put on its crank end dead center by the method given under (*a*) and the lead made equal to that shown by the diagram by swinging the eccentric on the shaft. This completes the setting, but the lead and port opening

should be measured on the head end as a check, and compared with the values given by the diagram.

(d) **Valve Setting by the Indicator.** The ultimate test by which the steam distribution must be judged is by the indicator. The valve setting should be such that not only does the engine run smoothly, but that it should give the maximum power for the steam used.

Valves may be set by deductions from the indicator diagrams, and the previously described methods dispensed with. The procedure may be more laborious and less systematic, since it is largely cut and try.

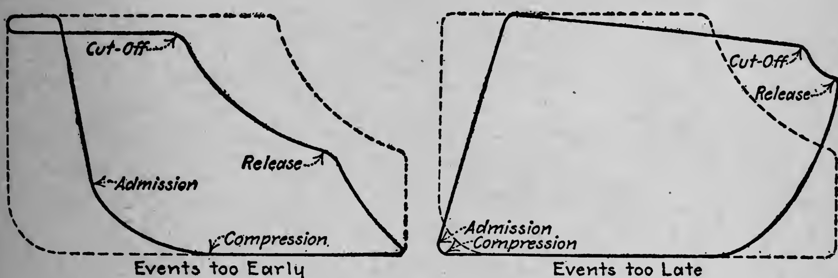


FIG. 89.—Faulty Valve Setting shown by Indicator Diagrams.

Fig. 89 shows indicator diagrams, superposed on ideal ones represented by dotted lines, from which may be noted the variations caused by too early and too late occurrence of the events of the stroke.

If the admission is too early (that is, if there is too much lead), full boiler pressure is admitted to the cylinder before the beginning of the stroke and the admission line rises at a point other than the extreme of the diagram. If the admission is too late (too little lead), the admission line slants inward because the full boiler pressure does not reach the cylinder until the piston has advanced appreciably in its forward stroke.

The cut-off may be too early on either end of the cylinder only in relation to that on the other end; they should be approximately equal for equal division of the load. The simple slide valve engine does not cut off earlier than six-tenths of the stroke; if the cut-off on either end is much later than this, as shown by the point at which the expansion line begins on the diagram, either the valve is not well designed or its setting is imperfect.

If release takes place too early, the pressure is released from the cylinder before the end of the stroke; the result is a curtailing of the expansion line and a sharp drop down to the backpressure line. If the release is too late, expansion may take place clear to the end of the stroke, but when the piston returns there is not enough opening for the steam to exhaust through freely. The result is excessive back pressure at the beginning of the return stroke as shown by the rounded toe of the diagram in Fig. 89.

The compressions on the two ends should begin at approximately the same percentage of the stroke. If there is too little compression, the engine will not run smoothly; if too much, the maximum power is reduced, although the economy in the use of steam may be increased. Compression tends to reduce the steam loss due to clearance.

To set a slide valve by the indicator, it is necessary to have diagrams from both ends of the cylinder. Faulty steam distribution may then be determined in detail by the effects on the diagrams as just described. A knowledge of the effects of the adjustments of stem length and eccentric position, as given in the table on page 210, then enables one to correct the steam distribution. The engine should then be indicated again, and further corrections made if necessary.

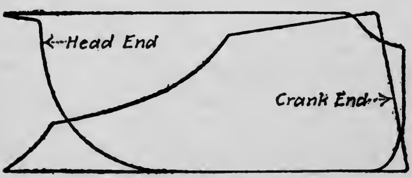


Fig. 90.

As an example of the reasoning involved, consider Fig. 90. It is well first to make a table showing the existing characteristics of the diagrams, as follows:

	Admission.	Cut-off.	Release.	Compression.
Head end....................	Early	Late	Late	Early
Crank end...................	Late	Early	Early	Late

By reference to the table on page 210, it is seen that to change the angular advance will aggravate some of the errors, which-

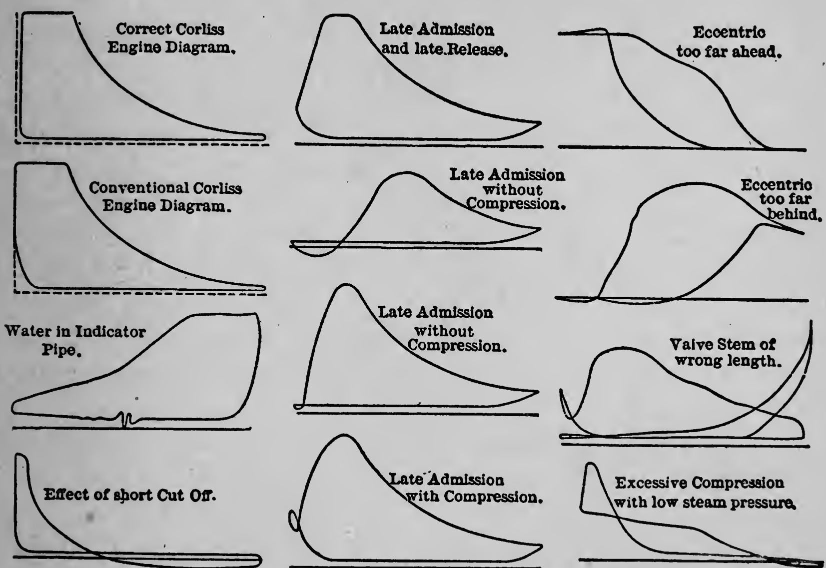


FIG. 91.—Faulty Valve Setting and Diagrams.
(Reproduced from *Power*.)

ever way it is adjusted. But if the stem length is increased, the events will tend to equalize and change to proper amounts. The stem length should therefore be increased and diagrams taken, the procedure being repeated until the cut-offs are about equal. It may then develop that all of the events are too

early or too late, in which case the eccentric should be adjusted until the admission lines are vertical. If the release and compression are then satisfactory, the setting is acceptable. Samples of indicator diagrams showing faulty valve setting, etc., are shown by Fig. 91.

(e) **Study of the Steam Distribution by the Bilgram or Zeuner Diagrams.** This may be done before or after setting the valve in order to decide upon the best setting or how the setting may be improved, thus possibly saving a number of trial settings and of indicator trials. For this purpose, the necessary measurements are made of the valve and engine parts, from which a number of diagrams are constructed with different adjustments of eccentric and stem length assumed. Indicator diagrams may be constructed from the data yielded by the Bilgram or Zeuner diagrams (the expansion and compression curves being drawn as rectangular hyperbolas), by comparison of which, the best setting may be selected.

The measurements to be made from the engine are of the connecting rod and crank lengths, the eccentricity, and the lead and outside and inside lap on one end. The outside lap at either end may be readily determined by noting the distance the valve must move from mid-position to begin to uncover the port. The mid-position may be found by making a prick point or scratch on the valve stem and measuring to this mark. The inside lap at either end may be obtained by subtracting from the length of the valve face, the length of the port and the outside lap, the dimensions being taken from the end considered. The sum of the outside laps should also be found as described under (*c*), and the sum of the inside laps, found similarly, by subtracting the distance between the inside edges of the valve faces from the distance between the inside edges of the ports.

The Bilgram diagram may now be laid out by drawing first the eccentric circle and the lead at the end for which it has been measured, say the head. (See Fig. 87.) Next, the head end

lap circle is drawn with its center at such a point, x, on the eccentric circle as to make the lap circle tangent to the lead line. The angular advance line is then drawn in, which locates the centers of the other lap circles. Having described these the events of the stroke, in percentages, may be determined.

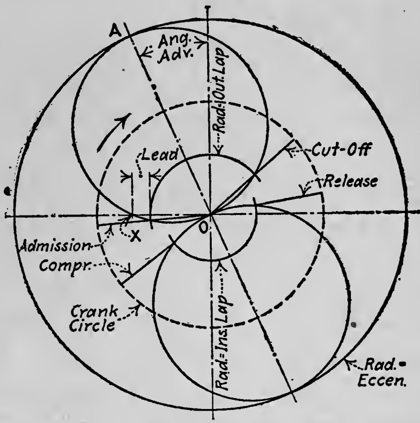


FIG. 92.—Zeuner Valve Diagram.

The Zeuner diagram may be laid out similarly, starting with the eccentric circle (see Fig. 92). A point X is then located at a distance from the center of the eccentric circle equal to the lap plus the lead on the head end. The circle OXA is next constructed, which fixes the angular advance by the line OA. When the lap circles are put in, the events of the stroke may be determined in the usual manner.

It may be noted that it is convenient to work with the sum of the outside laps and of the inside laps, since, if it is desired to reconstruct the diagram with a different lap on one end, the opposite one is readily determined by subtraction.

Problem 42$_1$. Verify the table on page 210 by the method given in the text following it.

Problem 42$_2$. The lead on the head end of a steam engine is found to be $\frac{1}{16}$ in., and on the crank end $\frac{3}{16}$ in. If the leads are to be made equal to $\frac{1}{16}$ in., should the valve stem be lengthened or shortened and how much? Should the angular advance be increased or decreased?

Problem 42$_3$. A D-slide valve has a face 1.5 ins. long at each end, and the distance between the faces is 2.75 ins. The ports are each $\frac{11}{16}$ in., and the outside edges of the ports are 4.50 ins. apart. If the head end outside lap is .625 in., what are the values of the other three laps?

Ans., $\frac{5}{8}$, $\frac{3}{16}$, $\frac{3}{16}$ in.

Problem 42_4. The valve in the preceding problem has an eccentricity of 1.375 ins. Find the lead and the maximum port opening on the crank end for equal cut-offs, using the Bilgram diagram. The connecting rod is 28 ins. long and the crank 5 ins.

Problem 42_5. Find the events of the stroke in per cents, using the data of the last problem and the Bilgram diagram.

Problem 42_6. Repeat the last problem, using the Zeuner diagram.

Problem 42_7. Why cannot a valve be set for equal leads and equal cut-offs at the same time? If for equal leads, which cut-off is greater?

Problem 42_8. How could the events of the stroke be measured direct from an engine whose valve was set, by reference to the valve and measurements of the cross-head travel?

Problem 42_9. Set the valve of an engine for equal cut-offs. Then measure the events of the stroke from the engine, from the indicator diagram actually obtained, and from the Bilgram diagram.

Test 43. Setting a Corliss Valve Gear

Principles. The action of the Corliss valve gear is explained in numerous works on the steam engine. The student should have an understanding of this action to grasp the following:

The setting may be considered in two parts; first, for the proper laps and leads, and, second, for the governor regulation. Usual values of the laps and leads are as follows, the smaller ones corresponding to the smaller engine sizes.

Steam lap........................	$\frac{1}{32}$ to $\frac{1}{2}$ inch.
Exhaust lap........................	$\frac{1}{16}$ to $\frac{3}{16}$ inch.
Lead........................	$\frac{1}{32}$ to $\frac{1}{8}$ inch.

The laps are adjusted and measured when the linkage is in its mid-position, as indicated by Fig. 93. This figure also names the terms by which the various parts will be referred to, so that the following instructions may be understood.

(a) **Adjustment of Laps and Leads.** The first step is to set the wrist plate in its center position, the hook rod being lifted to free the wrist plate. There is generally a mark on the wrist plate hub which registers with three stationary marks, as shown in

Fig. 93. The middle stationary mark indicates the correct center position of the wrist plate, and the outer marks, its extreme positions. If desired, the center position may be checked with a plumb line. Having located this, the hook rod length should be adjusted so that, when engaged with the wrist plate, the rocker arm stands vertical as shown by a plumb line. The eccentric

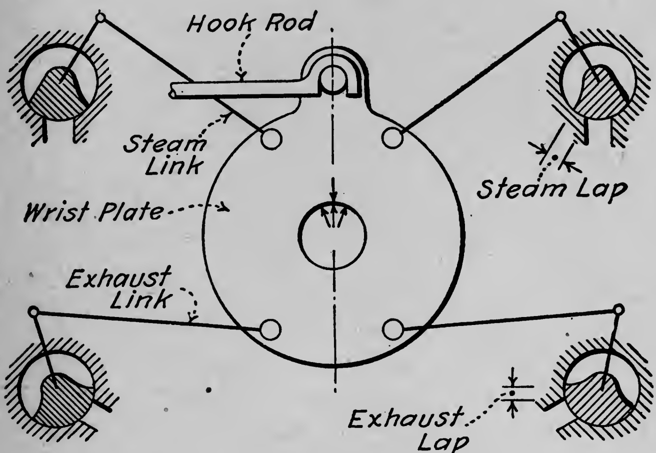


Fig. 93.

rod length may now be adjusted so that the eccentric throws the wrist plate to the correct extreme positions.

Now, with the wrist plate in its center position and the steam valves hooked up, the laps may be made the desired amounts by altering the lengths of the steam and exhaust links. The laps are readily measured when the exhaust bonnets are removed, as lines will be found to indicate the edges of the valves and ports. This accomplished, put the engine on head end dead center (see Test 42 (*a*) for method) and make the head end lead the selected value by swinging the eccentric on the shaft, thus moving the valve to the desired position for lead. Then fasten the

eccentric, put the engine on the other dead center, and measure the crank end lead. If the leads are not equal, the crank and steam link may be changed in length slightly to obtain equality.

(b) **Adjustment of Governor Rods.** With the governor in the lowest running plane (but not in the safety stop plane), turn the flywheel by hand until the wrist-plate is at its limit of travel towards the head end as shown by the mark on the wrist-plate hub. The crank-end governor rod, having been previously lengthened, the crank-end valve will not trip during the motion. This rod is now slowly and carefully shortened until tripping occurs, the wrist-plate being stationary at the head end limit during the adjustment. This done, the head end rod is dealt with similarly. The cut-offs will then be latest when the governor is in its lowest running plane.

The no-load action of the governor should next be checked. Block up the governor to its highest plane, in which position the steam valves should just fail to hook up.

Another procedure is to set the governor rods at the highest position of the governor so as to open the valves a very small amount when the valves are tripped; and, for the lowest position the valves are not released at all. Note that a setting at one limit of the governor travel cannot be made without affecting that at the other.

(c) **Adjustment of Dash-pot Rod.** Each steam valve hook engages with a "catch-block" on an arm rigidly fastened to the valve, and this arm is connected with the dash-pot rod. By lengthening or shortening the dash-pot rod, the catch-block may be raised or lowered. When the valve hook is in its lowest position (that is, when the wrist plate is at an extreme position), the corresponding dash-pot rod should be changed in length until there is an equal clearance above and below the catch-block, between it and the hook.

(d) **A Check of the Setting by Indicator** should be made after the completion of all adjustments. In particular, the amount of compression should be ascertained, and the action of the dash-pots

towards a sharp cut-off, together with the equalization of cut-offs. Improvements in the setting may be made often by changing the steam and exhaust links.

Problem 43₁. Will increasing the length of a steam link increase or decrease the lap and the lead? Why? What effect will an increase of the angular advance have on laps and leads? Why?

Problem 43₂. If an indicator diagram shows the compression on the head end to be too early, should the exhaust link length be increased or decreased, and what effect will the readjustment have on the release?

44. The Mechanical Efficiency Test of a Steam Engine

Principles. The mechanical efficiency of a steam engine is equal to the useful horse-power divided by the horse-power developed by the steam in the cylinder. If a Prony brake is used to measure the useful horse-power, then (see Test 6)

$$\text{B.h.p.} = BWN$$

is the useful horse-power.

Using the following notation,

I.h.p. = Indicated horse-power;

P_h and P_c = the mean effective pressure on the head end and crank end respectively;

L = the length of the stroke in feet;

a = the area of the cylinder, in square inches;

a' = the area of piston rod, in square inches;

N = the number of revolutions per minute;

then the cylinder horse-power may be expressed,

$$\text{On the head end, I.h.p.}_h = \frac{P_h LaN}{33{,}000} = K_h P_h N. \quad . \quad . \quad (1)$$

$$\text{On the crank end, I.h.p.}_c = \frac{P_c L(a-a')N}{33{,}000} = K_c P_c N \qquad (2)$$

K_h and K_c are called the "engine constants."

$$\text{Total I.h.p.} = \text{I.h.p.}_h + \text{I.h.p.}_c.$$

Mechanical friction causes a loss of power so that all the work developed in the cylinder does not appear at the flywheel. The friction horse-power equals

$$\text{F.h.p.} = \text{I.h.p.} - \text{B.h.p.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

When the brake is entirely removed from the engine, the work done in the cylinder to keep the engine running is expended to overcome friction only.

The function of the engine governor is to keep the speed constant. This is done by the action of any change of speed, through centrifugal force, to alter the position of the governor weight. More or less steam is admitted into the cylinder according to the position of this weight, and therefore more or less cylinder work is done. To accommodate itself to a variable load, then, the engine *must* allow some change of speed to regulate the cylinder work. At no load, with the governor admitting the least steam, the speed is greatest, and at maximum load, least. Good regulation requires that the difference between these limits of speed be small.

For comparative purposes the speed regulation is expressed as a percentage variation from the mean speed. If N_m and N_n are the speeds at maximum and no load, respectively, then

$$\text{Per cent speed regulation} = \frac{(N_n - N_m) \div 2}{(N_n + N_m) \div 2} \times 100.$$

(a) **The Indicated Horse-power** is determined by measuring the quantities in formulas 1 and 2, the engine constants being calculated before the test. The mean effective pressures are found from indicator diagrams by integration with a planimeter. The speed is determined with a hand or continuous counter.

The brake horse-power being the independent variable, it is first calculated what net forces should be obtained at the brake scales to produce the required horse-powers. The engine

is then operated at one of these horse-powers by adjusting the tension of the brake strap to give the necessary force at the scales, and to maintain it at that value. This means constant observation of the brake scales and occasional regulation of the brake-strap tension. The engine should be operated at each load for ten minutes or more so that it may adjust itself to the new conditions of load and friction. A number of indicator diagrams and measurements of the R.p.m. are then taken. It is convenient to average the readings of P_h, P_c, and N at each load and to use the averages for substitution in formulas 1 and 2.

A more convenient, although approximate, formula for calculating the indicated horse-power, is as follows:

$$\text{I.h.p.} = k(P_h + P_c)N$$

in which k is the average of the two engine constants or $L\left(a - \frac{a'}{2}\right)$ $\div 33{,}000$. A brief consideration will show that the more nearly is the area on the crank end equal to that on the head (that is, the smaller is the area of the piston rod relative to that of the cylinder) the less error is there in the approximate formula. Also, the less difference is there between the mean effective pressures on the two sides of the piston, the less error is there. When they are equal the results from the two methods of calculation are the same.

The following is a rational formula by which the per cent of error ensuing from the approximate formula may be determined for any conditions. Let

R = Ratio of P_c to P_h;
d = Diameter of the piston rod, inches;
D = Diameter of the cylinder bore, inches.

Then

$$\text{Per cent of error} = 50 \times \frac{d^2}{D^2} \times \frac{1-R}{1+R}.$$

Using this formula, it will be found with usual proportions of d and D, that the mean effective pressure on one end must be two or three times that on the other to produce more than one per cent of error. It is thus clear that the approximate calculation may almost always be used, since any engine with properly set valve would give a more uniform division of the load than this under normal conditions. Some engines divide the cylinder work quite unequally at light loads; in such a case the formula will show whether or not the correct method should be used. A convenient rule is as follows:

If the mean effective pressure on one end is not greater than

$$\frac{50d^2+D^2}{50d^2-D^2},$$

times the mean effective pressure on the other, then the error is less than 1 per cent.

(b) **The Friction Horse-power** at each load is found as indicated by formula 3. Tests have shown that this quantity is very nearly constant at all loads of a given engine, if properly operated. As the conditions of lubrication affect friction and therefore mechanical efficiency, the lubrication should be given special attention and maintained uniform throughout the test. It is advisable to plot a curve of F.h.p. vs. B.h.p. as the test progresses as this is closely indicative of the accuracy of the results.

(c) **The Efficiency is** found by dividing each value of the B.h.p. by the corresponding value of the I.h.p.

(d) **Speed Regulation.** Values of N_m and N_n have been obtained for the other results, and these may be used to get the speed regulation. Another test consists in quickly throwing the entire load on or off and noting the resulting momentary variation in speed. This is greater than that produced by a gradual change of the load, and it should be determined with a chronograph or tachagraph.

Any change in boiler pressure will affect the speed regulation, so this should be kept constant during the test if possible, and any variations noted.

Problem 44_1. If the F.h.p. is constant at all values of the B.h.p., deduce the forms of the curves of B.h.p. vs. I.h.p. and efficiency vs. B.h.p. Will these curves pass through the origin or not? Why?

Problem 44_2. Figure what may be the maximum ratio of mean effective pressures with a 14-in.×30-in. engine having a $2\frac{7}{16}$-inch piston rod, so that the error from using the approximate formula for I.h.p. will be less than 1 per cent. *Ans.*, 5, nearly.

Problem 44_3. Calculate the length of the adjustable arm of a polar planimeter, so that horse-power may be read direct when indicating the engine in the preceding problem. (See Test 14 (*d*).) The diameter of the record wheel is 0.79 in. and there are 100 graduations on the wheel. One graduation to equal 1 H.p. Length of indicator diagram = 5 in. Spring scale = 60. Average value of N is 80. *Ans.*, 3.62 in.

Problem 44_4. A 50-horse-power engine running at 200 R.p.m. is tested with a brake with an 8-ft. arm. If its unbalanced weight is 10 lbs. and the weight of the pedestal by which its thrust is transmitted to a platform scales is 5 lbs., what should be the scale readings for an efficiency test of six runs? *Ans.*, 179 lbs., max.

Problem 44_5. Run a series of tests on a steam engine to show the relation between steam pressure at admission if the governor is throttling, or cut-offs, if of the cut-off type, under variable brake horse-power.

45. * Economy Test of a Steam Engine

Principles. Economy tests on steam engines are made to determine the amount of steam and heat they consume per unit of power, under different conditions. The most important variable in such tests is the brake horse-power; it is usual to vary it throughout the working range for a complete economy test.

Economy results should always be based on the brake horse-power, but the indicated horse-power is often used because it is inconvenient or impossible to brake the engine. The unit "horse-power-hour" will be used, meaning the amount of work developed in one hour by one horse-power. The heat equivalent

* See also Appendix B, items 17 and 36.

of this work should be remembered. Since 33,000 foot-pounds of work are developed in one minute by one horse-power and 778 foot-pounds equal one B.t.u., then in one hour

$$\frac{33{,}000}{778}\times 60 = 2545 \text{ B.t.u.} = 1 \text{ H.p.-hr.}$$

The steam supplied to the engine is generally expressed in pounds per horse-power-hour, that is, the pounds of steam supplied in one hour divided by the horse-power. The result should be considered in connection with the condition of the steam, since less will be needed at high pressure or superheat, and more if it is wet. It is therefore necessary to include a statement of the condition of the steam in the expression for steam consumption.

The heat consumed by the engine may be figured from the total weight of steam supplied and the heat *consumed* from each pound. This latter expression requires consideration. Each pound of steam received by the engine carries heat in the form of latent heat and heat of the liquid. Part of this total heat is converted into useful work and part is lost to friction, radiation, etc., in the cylinder; the balance is rejected. During passage through the cylinder, part of each pound of H_2O originally supplied is condensed, so that when it appears in the exhaust, it is partly water and partly steam at a considerably reduced pressure. The heat content of such a mixture is $h'+x'L'$ (see page 140 for notation), so that the heat consumed by the engine might be judged to be $H-(h'+x'L')$, in which H is the total heat of the steam supplied, allowing for wetness or superheat, if any. That is, the heat consumed is the difference between the heat supplied and the heat rejected. Of the heat rejected, however, the latent heat cannot be made use of without bringing in auxiliary apparatus such as a feed-water heater. Therefore it is reasonable to charge the latent heat of the exhaust against the engine. On the other hand, the heat of the liquid of the exhaust may be reclaimed if the

exhaust is condensed and returned to the boiler. Therefore, a fair, though arbitrary, standard is

$$H-h'$$

for the heat consumed per pound of steam. Under ideal conditions the exhaust could be returned to the boiler as feed water without any drop of temperature; h' may therefore be taken as the heat of the liquid corresponding to the pressure or temperature of the exhaust.

If S is the weight of steam supplied per horse-power-hour in pounds, then the heat consumed per horse-power-hour is

$$S(H-h'),$$

in which S is the weight of steam per horse-power-hour including moisture, if wet; H is the total heat of the steam near the throttle valve of the engine, and equals $h+xL$, $h+L$, or $h+L+Cp(T-t)$, depending upon whether the steam is wet, dry, or superheated; and h' is the heat of the liquid corresponding to the pressure in the exhaust pipe close to the engine.

For future purposes it is well to note here that *the heat converted into work per pound of steam* is

$$2545 \div S,$$

S being based on the indicated horsepower.

(a) **Steam Consumption.** Having measured the horse-power, it is only necessary to get the hourly rate of steam supplied. This may be done in various ways as follows.

By Indicator Diagram. The engine itself makes a crude form of steam meter, since at every stroke a definite volume of steam is taken into the cylinder which can be measured from the indicator diagram. Referring to Fig. 32, page 51, at the point b cut-off takes place, the cylinder is closed to the boiler, and the cycle commences with a volume of steam equal to that repre-

sented by the point b. This comprises the clearance volume and the volume of the piston displacement up to the point b, or $(c+C)D$ cubic feet; c being the clearance expressed as a part of the piston displacement; C the cut-off expressed as part of the stroke, and D the piston displacement in cubic feet. Assuming the steam having this volume to be saturated, we may find its density in pounds per cubic foot from its pressure. Calling the density W, we have $W(c+C)D$ lbs. as the weight of the steam at the point b. Not all of this steam has been furnished by the boiler at the beginning of the cycle, however, since some steam from the previous stroke was compressed in the clearance space. Just before the valve opened to the boiler, the steam in the cylinder had a pressure and volume corresponding to the point e, Fig. 32. Its weight then is wcD, in which w is the density of saturated steam at the pressure at e. The amount of steam furnished by the boiler to one end each revolution is therefore $W(c+C)D-wcD$ lbs. If N is the revolutions per minute, the weight per hour is

$$60\,ND\{W(c+C)-wc\}.*$$

This is the weight used on one end of the cylinder on the assumption that the steam is saturated just after cut-off. The weight on the other end may be found similarly. Now, the steam is *not* dry because of initial condensation, and it may have varying degrees of wetness depending upon the cut-off, the working range of temperatures, type of engine, etc. Numerous empirical formulas have been proposed for the calculation of cylinder condensation, an important item, since for simple engines it amounts to between 20 and 50 per cent of the amount of steam shown by the diagram. A method has been proposed by Professor J. Paul Clayton which takes advantage of the law discovered by him that there is a definite relation between the amount of cylinder condensation and the exponent of expansion

* See also foot-note, page 238.

in the equation PV^n = a constant, at all events for certain types of engines. (See Journal A.S.M.E., April, 1912.) He plots the indicator diagram on logarithmic coordinates (see Test 41 (*c*)) and from the slant of the expansion line gets the value of n. By using a set of curves giving the quality of the steam in the cylinder just after cut-off at various values of pressure and n, the quality of the steam at cut-off and the cylinder condensation may be calculated. The method takes no account of leakage of steam past the valve, which may never enter the cylinder, but is nevertheless consumed by the engine. Since its publication the method has not found favor among practicing engineers and therefore is probably not generally applicable.

The clearance should be determined as described under Test 41. The expression for cut-off, C, may be found by dividing the distance of b, Fig. 32, from ae, by the length of the diagram, in inches.

For the sampling of diagrams, see Test 10 (*e*).

By Condenser. All of the steam used is passed into a surface condenser, and the condensate weighed for a counted time. This is probably the most accurate way of testing for steam consumption. *Numerous time-quantity readings should be made to determine uniformity of flow.* The condenser must be operated with more circulating water than is used in practice, so that the condensate will emerge cool; otherwise a large amount may escape unweighed by evaporation. The piping between the condenser and engine must be examined for tightness, and the condenser tested for leakage. The latter may be done by running the air-pump when no steam is flowing, and noting if any circulating water is drawn through, preferably when the condenser is hot. If it leaks a small amount, a leakage rate may be found and applied as a correction to the results. If the engine is to be tested "non-condensing," that is, with no vacuum, the condenser must be vented in order to establish this condition.

By Steam Meter. One of the commercial forms of steam meter may be used in the steam pipe supplying the engine. See Test 30.

By Feed-water Measurement. For this method, the engine and boiler supplying it should be isolated so that all of the steam generated by the boiler is used in the engine. Sometimes it is necessary to supply steam to auxiliary apparatus, such as a feed pump, from the boiler supplying the engine tested. In such a case, it is necessary to make a separate measurement of this steam, either by establishing the rate beforehand or, better, by condensing the steam used by the auxiliary during the test. It is necessary to examine the boiler and piping for tightness, the latter especially at branches stopped by valves. This is done by closing all valves in branches and the main stop valve at the engine so that the supply pipe is open from the boiler to the engine valve, but closed everywhere else. With a quiet furnace fire so that there is no active evaporation, the level is then noted in the water column at a number of uniform intervals of time. If the level falls, leakage is taking place, and the rate should be determined from the area of water surface calculated from the measurements of the boiler. This leakage may then be applied to the results of a test as a correction.

The feed water may be measured by any of the methods given under Test 54.

Willan's Law states that the weight of steam per unit of time used by an engine with a throttling governor varies directly as the indicated horse-power of the engine, very nearly. The work represented by an indicator diagram in which the expansion follows the law that $PV=$ a constant (very nearly the case with steam) is mathematically proportional to the initial pressure, cut-off being constant. As the pressure is proportional to the density of the steam, approximately, it follows that the weight of the steam is proportional to the work, and the indicated horse-power. The weight of steam used also varies directly with the

brake horse-power, since B.h.p. = I.h.p.—a constant. Experiment has shown that this relation applies not only to throttling engines, but to those of the cut-off type, and to steam turbines.

The practical application of Willan's law lies in the consequence that if the weight of steam used per hour by an engine is plotted against its indicated or brake horse-power, the result is a straight line. This furnishes a check upon the results of a test as the test proceeds; if points so plotted do not follow a straight line, there is error. It should be noted that at overloads, the curve is apt to deviate slightly from straightness, and lean toward the axis of steam weights.

The duration of the test should depend upon the uniformity of conditions, the capacity of the engine, and the method used for measurement of the steam. During each test at a given load, weighings are made at uniform time intervals, say ten or fifteen minutes. When there are four to six of these of nearly equal amounts recorded consecutively, the run may be discontinued, provided the error of starting and stopping is within reasonable limits. (See page 9.) This error depends upon the method of measuring the steam. If a condenser is used, the error will be relatively small, since it equals the difference in the amounts of condensate in the condenser at starting and stopping. If the boiler method is used, the test must be much longer, as there may be large error in the level of the water column, due to differences of density, ebullition, etc. With a steam meter, the test need be only long enough to get sufficient indicator diagrams for a fair average, provided the engine has been previously given a settling run.

Determinations of the condition of the steam supplied as to quality and pressure, and of the pressure or temperature of the exhaust, should be made and included in the statement of steam consumption per horse-power-hour.

The relation between the steam consumption in pounds per

indicated horse-power-hour, S_1, and per brake horse-power-hour, S_2, is

$$S_1 = S_2 \times \text{mechanical efficiency.}$$

(b) Cylinder Condensation is often figured by subtracting the total steam used per hour, as shown by the indicator diagram, from that determined by direct measurement. The difference includes not only cylinder condensation, but valve leakage in the case of engines which do not have separate valves controlling the exhaust.

(c) Thermal Efficiency. There are two standards, one having a commercial, the other a scientific value. The former is the ratio of the heat equivalent of the useful work to the heat units consumed as defined under "principles." Since there are $S(H-h')$ heat units consumed for every horse-power-hour of useful work, and since the heat equivalent of a horse-power-hour is 2545 B.t.u.,

$$E = \text{thermal efficiency} = \frac{2545}{S(H-h')},$$

in which S may be based on indicated or brake horse-power. If the former, the result is the "cylinder efficiency"; if the latter, it is the "over-all efficiency."

H may be obtained from the pressure and quality determinations of the steam supplied, by use of the steam tables; h' is determined similarly from readings of a pressure gage or thermometer in the exhaust.

Efficiency Ratio. This is the ratio of the actual engine efficiency (as defined above) to that of an ideal engine working without clearance space and with non-heat-conducting cylinder walls with cut-off early enough to allow expansion clear down to the back pressure. There will then be no clearance loss and the expansion will be adiabatic. The indicator diagram representing these conditions is shown by Fig. 94, and is known as the Rankine cycle, although it is

by some writers attributed to Clausius. The initial pressure and quality of the steam and the back pressure are assumed to be the same in the ideal engine as in the actual. The "efficiency ratio" is a more reasonable standard for efficiency than the commercial standard since the latter charges against the engine heat that it could not use under even ideal circumstances because of the limitations in the operation of the working medium.

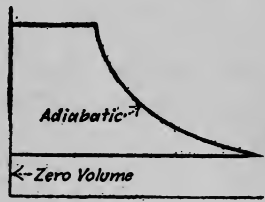


FIG. 94.—Rankine Cycle.

The ideal *cycle* efficiency is the heat available for work, per pound of H_2O, divided by the heat added per pound. Under the conditions of the assumed ideal engine the heat converted into work is the difference between the total heat H of the steam supplied and the total heat of the exhaust H', since there are no losses. H' is the theoretical amount of heat left in a pound of steam after it has expanded adiabatically from the given pressure and condition as to quality to the given back pressure. It may be calculated by the use of entropy tables, but it is more convenient to get it directly from the Mollier total heat-entropy diagram which gives the heat of steam under all conditions and at various stages of adiabatic expansion. (See Example 3, p. 378.)

Since the heat added per pound is the same in the ideal cycle as for the actual engine as previously defined. the *cycle* efficiency is

$$(H-H') \div (H-h').$$

Then the

$$\text{Rankine Eff. ratio} = E_c = \frac{2545}{S(H-h')} \div \frac{H-H'}{H-h'} = \frac{2545}{S(H-H')}.$$

Since $2545 \div (H-H')$ is the steam consumption, S_c, in lbs. per H.p.-hr. of the ideal engine

$$E_c = \frac{2545}{H-H'} \times \frac{1}{S} = \frac{S_c}{S}.$$

This is a useful form, since by it the cylinder or over-all efficiency may be readily obtained, depending upon the basis of S.

Problem 45_1. An indicator diagram shows the cut-off to be one-fifth of the stroke. The pressure at cut-off is 110 lbs. abs.; at the end of compression it is 25 lbs. abs. The engine is 10 in. × 12 in. × 100 R.p.m., single acting with 4 per cent clearance. How many pounds of steam are supplied per hour, allowing 25 per cent of that shown by the diagram for condensation? *Ans.*, 233.

Problem 45_2. If the mean-effective pressure in the preceding is 25 lbs., what is the steam consumption in pounds per I.h.p.-hour? *Ans.*, 39.2.

Problem 45_3. If the steam in the supply pipe (last problem) is at 105 lbs. gage and contains 3 per cent of moisture, and if the exhaust is at 3 lbs. gage, what is the thermal efficiency? *Ans.*, 6.69%.

Problem 45_4. What is the Rankine efficiency for the foregoing? *Ans.*, 48.1%.

Problem 45_5. Deduce from the typical form of Willan's line, the form of the curve between steam consumption in pounds per B.h.p.-hour and B.h.p. What is the value of the steam consumption when B.h.p. = 0?

46. Test of a Multiple Expansion Engine

Principles. The results to be sought are in part identical to those for a simple engine; hence the principles under Tests 44 and 45 are appropriate and should be read in this connection. In addition, there are other data useful to the study of multiple expansion, principally pertaining to the indicator diagrams. The **sampling** of diagrams is therefore very important, and should be done according to Test 10 (*e*).

Although, for brevity, a compound engine only will be considered here, the methods apply equally to any multiple expansion.

(a) **I.h.p., B.h.p., F.h.p., and Mechanical Efficiency** may be found as for a simple engine, but in many cases the unit will be too large to brake conveniently, or a generator connection will make that procedure impossible.

In the former case, the F.h.p. may be determined by running the engine free and taking indicator diagrams, called under these circumstances "friction diagrams." The I.h.p. from such diagrams equals the friction horse-power. If the F.h.p. is assumed constant at all loads, this single determination of it at zero load enables the calculation of the B.h.p. at any load, since

$$\text{B.h.p.} = \text{I.h.p.} - \text{F.h.p.}$$

When there is a direct connected generator, the electrical load should be measured as for a steam turbine (Test 47 (*a*)). If the efficiency curve of the generator is known, the B.h.p. of the engine may then be closely estimated.

The I.h.p. may be obtained by figuring that for each cylinder separately, and adding them to get the total I.h.p. Or the "equivalent mean effective pressure" (to be defined later) may be used in a single calculation upon one cylinder.

Whenever possible, the B.h.p. should be used as the independent variable (if results from the engine alone are to be considered).

(b) **Steam Consumption** by condenser, steam meter, or feed-water measurement. See Test 45 (*a*).

(c) **Thermal Efficiency.** See Test 45 (*c*).

(d) **Equivalent and Aggregate M.e.p.** The equivalent M.e.p. referred to any cylinder may be defined as a pressure of such value as to produce the same horse-power in the referred to as in the actual cylinder. Thus,

Let A_h and A_l = net piston areas for high- and low-pressure cylinders, respectively.

P_h and P_l = the M.e.p.'s, high- and low-pressure cylinders, respectively.

Then the M.e.p. of the high-pressure cylinder, referred to the low, is

$$\frac{A_h}{A_l} \times P_h,$$

and the M.e.p. of the low-pressure cylinder, referred to the high, is

$$\frac{A_l}{A_h} \times P_l.$$

The combined or "aggregate" M.e.p. is one of such value that, if it prevailed in the cylinder referred to, there would be produced in that cylinder an I.h.p. equal to that actually produced by all cylinders. That is (calling the aggregate M.e.p., P_{hl}),

Referred to the H.P. cylinder, $P_{hl} = P_h + \frac{A_l}{A_h} \times P_l.$

Referred to the L.P. cylinder, $P_{lh} = P_h \times \frac{A_l}{A_h} + P_l.$

and similarly for three or more cylinders.

(e) Steam Accounted for by Indicator Diagrams. On p. 230 is deduced the relation that the weight of steam per hour shown by an indicator diagram equals *

$$60ND\{W(c+C) - wc\}.$$

Now, for a single cylinder engine,

$$\text{I.h.p.} = \frac{PLaN}{33{,}000} = \frac{PN(D144)}{33{,}000}.$$

* This formula is sometimes quoted with $w(c+k)$ in place of wc, $c+k$ then standing for the volume of the steam at the beginning of the compression curve.

Dividing the one equation by the other, we have

$$\text{Wt. of steam per H.p.-hour} = \frac{13,750}{P}\{W(c+C)-wc\},$$

in which P is the mean effective pressure, c and C are the clearance and cut-off volumes expressed as parts of the piston displacement, respectively; and W and w the densities of saturated steam at the pressures of cut-off and compression, respectively. This last equation is the more convenient form for the diagram water rate.

The procedure then is to measure C, W, and w on representative diagrams (c being known) from the cylinder considered. The aggregate M.e.p. referred to that cylinder (defined under (*d*)) is then calculated and taken as the value of P in the water rate formula. The result is the steam accounted for by the diagram from the cylinder considered. The procedure is repeated for the other cylinders.

(f) **Combined Diagram.** Let

D_h and D_l = piston displacements, cu. ft., of high- and-low pressure cylinders, respectively.

v_h and v_l = clearance volumes, cu. ft., of high- and low-pressure cylinders, respectively.

Select representative diagrams from the cylinders, as in Figs. 95 and 96, and divide them into ten or more equal spaces by vertical lines.

Choose convenient scales of pressure and volume for the combined diagram, Fig 97, and lay them off on coordinate paper. Draw in the atmospheric pressure line.

Locate the vertical line 1–2, Fig. 97, at the volume graduation corresponding to the clearance volume, v_l. Locate the vertical 3–4 at the volume graduation corresponding to v_l+D_l.

Make ten (or more, corresponding to Fig. 95) spaces between lines 1–2 and 3–4 with equidistant vertical lines.

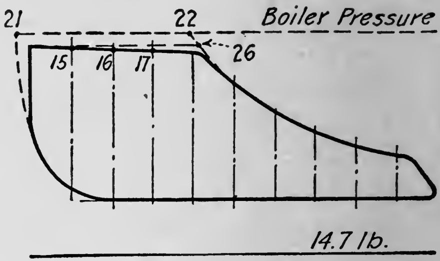


FIG. 96.—H.P. Diagram.

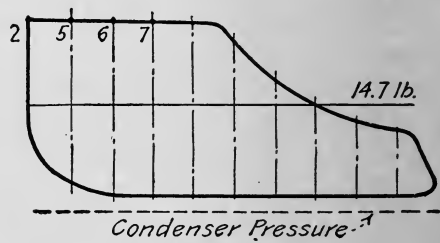


FIG. 95.—L.P. Diagram.

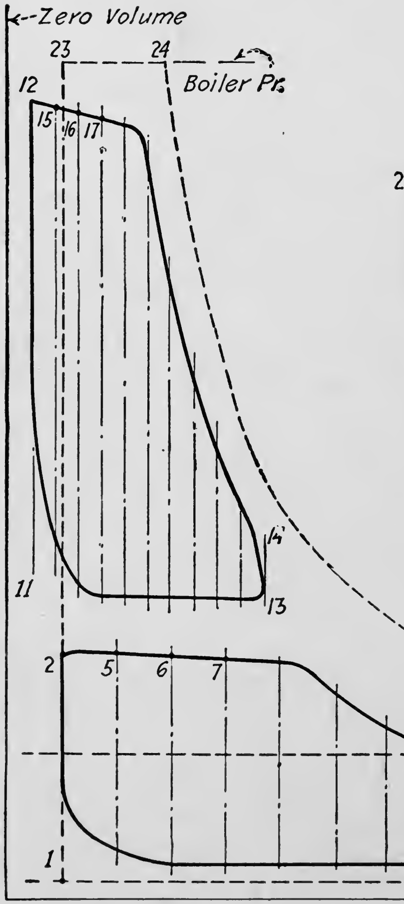
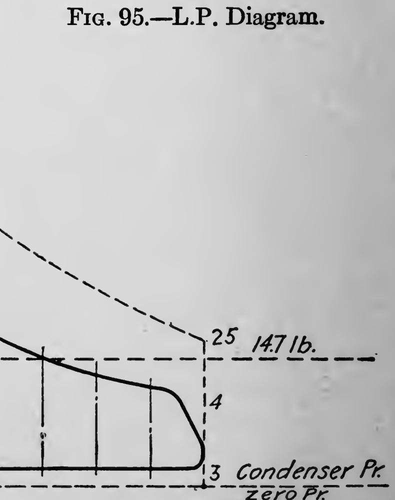



FIG. 97.—Combined Diagrams from a Compound Engine.

Points 5, 6, 7, etc., on the reconstructed diagram may now be located on these verticals by scaling the pressures of the corresponding points of Fig. 95 from the atmospheric line. In this way the full low-pressure diagram may be reproduced on the combined diagram.

Locate vertical 11–12, Fig. 97, to represent the clearance volume v_h; and 13–14 to represent $v_h + D_h$.

Draw equidistant verticals between 11–12 and 13–14, and locate points 15, 16, 17, etc., as for the low-pressure diagram. This establishes the high-pressure diagram.

The combined diagram, Fig. 97, fairly represents the whole expansion of the steam on a single P–V scale. When making comparisons with assumed ideal expansion diagrams, however, it should be borne in mind that Fig. 97 does not truly represent the continuous expansion of a given weight of steam. The high-pressure expansion curve is for all of the steam in the high-pressure cylinder; but the low-pressure expansion is for a lesser quantity of steam by the amount caught in the clearance of the high-pressure cylinder. Furthermore, valve and piston leakage will affect the validity of comparisons made with the hyperbolic expansion curve or the constant steam weight curves.

To draw in the ideal expansion line for the given conditions (the ideal being assumed hyperbolic, or $PV = \text{a constant}$), the boiler pressure line should first be drawn in on both the high-pressure and combined diagrams, as indicated on Figs. 96 and 97. The compression line of the high-pressure diagram (Fig. 96) is continued on a hyperbolic curve until it intersects the boiler pressure line at point 21. The expansion curve is continued similarly, thus locating point 22. The volume corresponding to the distance between 21 and 22 is now calculated in cubic feet. This volume is assumed to be that of the steam received from the boiler per stroke, as though in an engine without clearance.

Now continue line 1–2, Fig. 97, until it intercepts the boiler pressure line at 23. From 23 lay off 23–24 equal to the volume

just calculated. Through 24, construct the required hyperbolic curve by one of the well-known methods, using the line 1–2 and the condenser pressure line as zero axes.

(g) **The Ratio of Expansion** is equal to the volume of the low-pressure cylinder, including clearance, divided by the volume of the steam in the high-pressure cylinder, including clearance, at the point of cut-off. The last named is found as follows and is then termed the "commercial cut-off." Referring to the high-pressure diagram, Fig. 96, a horizontal is drawn through the highest point on the steam admission line. The intersection of this horizontal with the prolonged expansion curve, that is point 26, is the commercial cut-off.

(h) **The Diagram Factor** may be found by dividing the area of the combined diagram, Fig. 97, by the area of the ideal diagram, 23–24–25–4–3–1. Or it may be calculated by dividing the aggregate M.e.p. referred to the low-pressure cylinder by the ideal M.e.p. obtained from the following formula:

$$\text{Ideal M.e.p.} = \frac{P'}{R'}(1 + \log_e R') - p,$$

in which P' = boiler pressure, lbs. per sq. in., absolute;

p = condenser pressure, lbs. per. sq. in., absolute;

$$R' = \text{ideal ratio of expansion} = \frac{\text{length } 1\text{–}3}{\text{length } 23\text{–}24}.$$

47. * Economy Test of a Steam Turbine

Principles. The measurements and results, in general, are the same as those for a reciprocating engine, except that there can be no indicated horse-power determination, and often no brake measurements, an electrical load being considered instead. Since it is not possible to indicate a turbine, the term "internal horse-power" is sometimes used to express the equivalent of indicated horse-power, but it is not a definite quantity. It is

* See also Appendix B, items 19, 20, 37 and 38.

often impracticable to apply a brake to the turbine shaft on account of the direct connection of a generator; in such a case the turbine and generator must be tested as a single unit. For other details, see Test 45, principles.

(a) **Determination of the Useful Horse-power** may be made the same as for Test 44 if the turbine alone is tested with the use of a brake. Otherwise, the electrical load should be measured in kilowatts by taking either wattmeter readings or readings of amperes and volts. The horse-power equivalent to the electrical output, that is, the "electrical horse-power," may be found from

1 Electrical horse-power (E.h.p.) = 0.746 kilowatt.

For alternating current generators, the code of the A.S.M.E. states that the electrical output is the kilowatts delivered to the switchboard less that required for field excitation when the field is separately excited. Under this condition, therefore, measurements of the field current and voltage should be made.

(b) **Steam Consumption.** The pounds of steam consumed per hour may be measured by any of the methods described under Test 45 (*a*) except by indicator diagram. Dividing this by the kilowatts or horse-power gives the pounds of steam per kilowatt- or horse-power-hour.

It is often desirable, for purposes of comparison, to base the steam consumption on brake horse-power or "internal horse-power." The former quantity may be estimated, if an efficiency curve of the generator is available, by multiplying the steam consumption in pounds per electrical horse-power-hour at a given load by the generator efficiency at that load.

The internal horse-power is difficult to estimate with any degree of accuracy. It is sometimes assumed to be the quotient of the brake horse-power and the mechanical efficiency of a steam engine working under the same conditions. This, of course, is a crude estimate. Using it, however, the steam consumption may be expressed on the basis of internal horse-power.

For **duration of the test,** see page 233.

Additional Data. If there are traps arranged to catch condensation from the turbine casing, they should be drained regularly and the condensate weighed in with the steam consumed. Readings should be made of the pressure in the nozzle chamber to show the drop of pressure through the governor valve. If this drop is excessive, the steam consumption will be correspondingly high. The predetermined conditions of pressure, quality, and back pressure or vacuum should be kept as constant as possible, and the quantities carefully measured, as they have a decided influence upon the economy of a turbine. It is important that the barometer be read to get the required accuracy in the measurement of vacuum which should be expressed as an absolute pressure. This is because at low pressures the heat content of steam varies materially with the pressure, and also because turbine economy is more dependent upon the predetermined vacuum than pressure or superheat.

For the purpose of estimating the steam consumption of a turbine under a given set of conditions of steam pressure, quality and vacuum, such as are stated in the manufacturer's guarantee, the turbine being tested under somewhat different conditions, correction curves are supplied by the manufacturer. These show the number of pounds of steam per horse-power- or kilowatt-hour to be subtracted from the test result if the stated pressure is higher than the actual, the number to be subtracted if the stated superheat is higher, and the number to be subtracted if the stated back pressure is lower, or vice versa.

Willan's line should be plotted during all trials under variable output.

(c) **The Thermal Efficiencies** are obtained exactly as for Test 45 (*c*), the basis for the computation of useful work being the electrical, brake, or internal horse-power.

(d) **Separation of Losses.** The losses of a turbine may be considered in two groups. First, approximately constant losses

which include those due to friction and radiation Frictional resistance is encountered at the bearings and stuffing boxes, and between the discs and blades and the steam, and through windage of the external parts. Second, steam losses including leakage through clearance spaces, condensation, and faulty action of the blades and nozzles in not completely absorbing the energy of the steam.

These losses may not be separately measured without an involved test. Professor Carpenter has indicated a method of separating the two groups by the use of Willan's line. When plotted against brake horse-powers, this line intercepts the axis of steam consumption at a point which shows the steam consumed per hour to overcome the first group of losses, that is, at no load. The distance parallel to the load axis, measured in horse-power, necessary to move Willan's line so that it shall pass through the origin, equals the power given to the first group of losses. At any other load, this quantity is to be subtracted from the power that would be developed by the actual amount of steam consumed if operating on the Rankine cycle; the result is the actual brake horse-power plus the second group of losses, from which the second group may be readily obtained.

Problem 47_1. A turbine, tested at full load, consumes 3150 lbs. of steam in 90 minutes. The current delivered is 340 amperes and 220 volts. The generator efficiency at full load is 95 per cent and the friction losses of the turbine are 5 per cent of the power delivered to the generator. What is the steam consumption in pounds per kilowatt-hour, per E.h.p.-hour, per B.h.p.-hour, and per internal horse-power-hour? *Ans.*, 28.1 per Kw.-hr.

Problem 47_2. For the test of Problem 47_1, average values as follows were obtained. Steam pressure, 150 lbs. gage; superheat, 80° F.; vacuum, 20 inches mercury; barometer, 28.5 inches. What would have been the steam consumption if the following conditions held? Steam pressure, 180 lbs., abs.; superheat, 100° F.; vacuum, 28 ins.; barometer, 30 ins. Corrections are 0.05 lb. per kilowatt-hour for each pound difference in the steam pressure, 0.02 lb. for each degree of superheat, and 1.0 lb. for each inch of vacuum. *Ans.*, 20.5 lbs. per Kw.-hr.

Problem 47_3. What is the thermal efficiency for the results of Problem 47_1? How much efficiency should be added or subtracted for each unit of steam pressure, back pressure, and superheat. (See Problem 47_2.)

48. Economy Test of a Steam Power Plant

Principles. A complete test consists of measurements of the amounts of heat distributed and lost in the entire system, beginning with the energy of the fuel and ending with the energy delivered at the shaft or switchboard. For the present we shall consider only that part of the plant beyond the boiler, the subject of boiler trials being separately treated under Test 54.

The most important result from power plant tests is the fuel consumption in pounds of fuel per horse-power- or kilowatt-hour. Other results are the steam and heat consumption of the engine and the auxiliaries, and the heat balance. The latter is an equation between the total heat available and the various amounts accounted for in its distribution. They may be found in heat units per hour and expressed as per cents of the total heat per hour added to the feed water entering the boiler. If these percentages are separately multiplied by the boiler efficiency, the items of the heat balance will then be percentages of the heat available in the fuel. This follows from the fact that boiler efficiency is that part of the fuel energy which is added to the feed water and distributed as steam.

The boiler trial, which is part of the complete test and conducted at the same time, gives the efficiency of the boiler and its losses, all based on the heat value of the fuel. A complete heat balance is thereby obtainable.

The usual condensing power plant contains as auxiliaries a condenser, circulating pump, air pump, and feed water pump. There may also be a cooling tower and its auxiliaries, and a feed water heater using the exhaust steam from the auxiliaries, and such minor apparatus as separators, traps, etc., which may or may not be arranged to drain back to the boiler.

The rearrangement of the apparatus and piping for the purpose of testing should interfere as little as possible with normal operating conditions, and any departures from them that may

effect the results, such as different feed water temperatures, should be allowed for by making a separate test to measure the actual working quantities that have been altered.

Equipments of different power plants are too varied for very specific directions for the measurements of the various quantities. Any of the methods given under Test 45 (*a*) for the measurement of steam weights should be applied according to the exigencies of the particular test. Any one item of the heat balance may be omitted from the direct measurements, since one item may always be figured by subtraction.

The basis of all heat measurements is the heat contained by the feed water just before it enters the boiler or economizer if there is one.

Duration. If a boiler trial is included, its duration determines the length of the test; if the engine and auxiliaries only are to be tested, the duration should be the same as for a test of the engine alone. See Tests 45 and 54, principles.

(a) Steam and Heat Consumption of Engine and Auxiliaries per Horse-power- or Kilowatt-hour. Let w stand for weight of steam per hour; H, for its total heat per pound above 32 degrees depending upon its pressure and quality; and let h be the number of heat units per pound of the feed water above 32 degrees. Let the subscripts e, c, a, f, and b refer to the engine, circulating pump, air pump, feed pump, and boiler respectively. Then the steam consumption is

$$(w_e+w_c+w_a+w_f) \div \text{horse-power or kilowatts output.}$$

To get the heat consumption, the various values of H should be used for strict accuracy, as determined by the pressure and quality of the steam just before it is delivered to each unit, but in most cases the following form is practically correct.

Heat consumed per hour $= w_e(H_e-h)+(w_c+w_a+w_f)(H_b-h)$.

Dividing this by the horse-power or kilowatts gives the required quantity.

The heat consumed per hour by the engine and auxiliaries may also be determined by subtracting the heat lost to leakage and drips from the total heat added to the feed water (see (*b*) and (*c*) following).

The values of H should be obtained from the steam tables by reference to the measurements of pressure, temperature, and quality of the steam at the point considered. For H_b this should be immediately beyond the main stop valve of the boiler; for H_e, it should be just behind the main stop valve of the engine, between it and the separator if one is used. h is obtained from the temperature of the feed water just before it enters the boiler. If an injector is used, the temperature should be taken before entering the injector, since the heat added by that instrument comes from the boiler and returns to it. The heat consumed by the feed pump in this case should be figured separately and is very nearly equal to the heat equivalent of the work done in pumping. (See Test 52.)

(b) Heat Added per Hour to the Feed Water. If there is only one source of supply yielding W lbs. per hour, then the heat added is

$$W(H_b - h).$$

If in addition to this there is another supply such as would be obtained from traps, separators, etc., draining back to the boiler in an independent pipe carrying W_2 lbs. per hour, then the heat added is

$$W_1(H_b - h_1) + W_2(H_b - h_2),$$

the subscripts 1 and 2 referring to the main and auxiliary supplies, respectively.

The quantities H_b and h are measured as under (*a*). The weight of the main supply is generally measured by a weighing system as for boiler trials (see Test 54 (*b*)). Auxiliary supplies

of feed may be metered or directly weighed during a special test made for that purpose.

(c) Heat Lost to Leakage and Drains. Leakage may occur from the steam space of the boiler or from the steam pipes either to the air or to pipe branches, and it may occur below the water level in the boiler. Leakage which cannot be prevented should be measured by a preliminary test similar to that described on page 231 for leakage correction. It is assumed that the rate of leakage thus determined remains the same through the test of the plant. Calling it w_l pounds per hour, the heat lost is

$$w_l(H_b-h)$$

If the separators, traps, etc., do not drain back to the boiler, they may be kept closed during the test and drained at regular intervals into buckets of cold water whereby they are weighed. If w_d is the number of pounds of water withdrawn per hour, the loss is

$$w_d(H_b-h)$$

If this water is returned as an auxiliary supply, h_2 should be substituted for h in the above.

(d) Heat Balance. All the items, having been measured as above in heat units per hour, may now be expressed as percentages of the heat added to the feed water. Then the heat balance is

100% = % of heat consumed by engine
+ % of heat consumed by auxiliaries (stated separately or collectively)
+ % of heat lost to leakage and drains.

The first item on the right-hand side may be subdivided into the per cent of heat equivalent to the useful work and the per cents representing the various engine losses.

It should be noted that this method of analyzing the heat

distribution charges against the engine and auxiliaries the losses of radiation between the condenser and the boiler. Also, if a feed water heater is used, the heat consumed by the engine figures less, although the actual performance of the engine is exactly the same as if no heater were used, because the value of h, the heat of the feed water, is higher. The only way in which the effect of the heater appears numerically is in the heat added to the feed water, but no idea of the saving due to the heater may be formed.

Problem 48$_1$. A condensing plant is arranged so that the exhaust from the auxiliaries preheats the feed water after it leaves the condenser, in a closed heater. Tell how the plant could be tested so as to show the loss due to drop in the heat of the liquid between the engine exhaust and the heater entrance, and the gain due to the feed water heater.

49. Valve Setting of a Duplex (Steam) Pump

Principles. The duplex pump consists of two reciprocating steam pumps operating side by side and arranged so that the piston rod motion of one operates the steam valve of the other. Each pump consists of a water cylinder whose piston is directly driven by the piston rod of the steam cylinder.

The steam valves distribute the steam so that one pump is on its forward stroke when the other is on its return. The valves have neither laps nor lead, so that cut-off occurs simultaneously with release. The valve stem of each pump is driven by a rocker deriving its motion from the piston rod of the other pump (see Fig. 98). As each rocker moves through the whole piston rod stroke and as it is desirable that the valves throw only at the end of the stroke, a certain amount of lost motion is provided between the valve

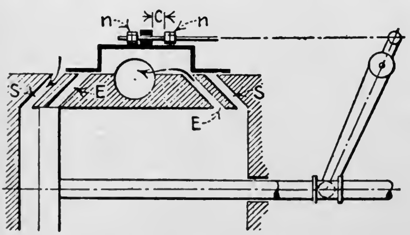


Fig. 98.—Duplex Pump Valve Motion.

stem and the valve. This is shown by the distance c in the figure. The valve stem passes freely through a hole in a lug on the valve, and gives motion to the valve only when the nuts n and n come into contact with the lug. By adjusting the lost motion, the valve may have more or less throw.

There are two steam and two exhaust ports for each steam cylinder, *SS* and *EE*. This is arranged as a safeguard against the piston hitting the cylinder head. If the valve should not close the exhaust port before the end of the exhaust stroke, the piston itself will close it, thus entraining steam in the clearance space which acts as a cushion.

(a) **Adjustment of Lost Motion of Valve.** This is accomplished by setting the valves in their mid-positions when the pistons are in their mid-positions.

The pistons should be set by reference to prick marks on the piston rods. The limiting positions of stroke should be measured by prying each piston rod with a bar to the extreme of its travel, pressure behind the pistons being released by opening drains or otherwise. When the piston rods are set midway between these positions, the rockers should be vertical.

With the steam chest cover removed, the valves are now placed so that they completely cover the ports. The nuts n, n, Fig. 98, are then adjusted so that there is an equal amount of clearance between each one and the valve lug. With some designs, there is only one nut to be adjusted on each stem, this nut then being between two lugs. In such a case the clearances may not be changed in amount, but only equalized. In other designs there is a " lost motion link " outside of the valve chest by which the clearances may be adjusted without opening the chest.

The proper amount of clearance depends upon the size and design of the pump and may be determined by trial. If the steam ports are to be opened full each stroke, then the total lost motion should be equal to the throw of the valve stem corresponding to

normal piston stroke less twice the steam port length parallel to the stroke.

(b) Readjustment after Trial. If, after the valves are set, the pump runs "lame," that is, the strokes are unequal in length or time, it may be because the resistances of the water cylinders are different. They should then be examined for leaky water valves, tight stuffing boxes, etc. Such faults being corrected, if the pump still runs lame, the lost motion of the valves must be readjusted until the desired equality of strokes is obtained. It need only be remembered that to diminish the lost motion makes the valve throw greater, thus providing better access to the steam and release to the exhaust.

Problem 49_1. Explain why the ordinary duplex pump cannot use steam expansively.

Problem 49_2. If the stroke of one piston rod of a duplex pump were curtailed at the head end, being normal in other particulars, what should be done to the valve motion to remedy it?

50. Mechanical Efficiency Test of a Reciprocating Steam Pump

Principles. The mechanical efficiency of a steam pump equals the useful or water horse-power, W.h.p., divided by the indicated horse-power of the steam cylinders. If D is the total head in feet pumped against, including suction and discharge, and W is the number of pounds of water delivered per minute, then,

$$\text{W.h.p.} = \frac{WD}{33,000}.$$

If there were no leakage of water in the pump from one side of the piston to the other or through the valves, then the work done per minute would be equal to the product of the total pressure pumped against in pounds per square foot and the volume

displaced by the piston in cubic feet per minute. If p stands for the pressure in the discharge pipe; p', for the vacuum in the suction pipe, both in pounds per square inch; and if L and A are the same as in the formula for indicated horse-power, and N is the number of working strokes per minute, then

$$\text{W.h.p.} = \frac{144(p+p')LAN}{33{,}000 \times 144}.$$

On account of leakage this formula is not a reliable one with which to determine the water horse-power. It may be used only by assuming a value for the leakage, and this may be very inexact.

Leakage of water in a pump, of the kind referred to, is called "slip." It is generally expressed numerically as a percentage of the piston displacement.

(a) Steam End Indicated Horse-power at Various Water Horse-powers. The indicated horse-power is found exactly as for a steam engine (see Test 44 (*a*)), except in regard to measurement of the stroke length, L. With direct acting reciprocating pumps, there is no constant limit to the stroke length and in ordinary operation it may vary materially. An average value should therefore be obtained from a number of measurements throughout the test. There is on the market a continuous recording device for this purpose, but if one is not available, the average stroke may be obtained from the average lengths of the indicator diagrams, the ratio of reduction being known. Another method is to set a scale against a mark on some projecting part of the piston rod, and to note the travel of this mark at regular time intervals.

The water horse-power may be varied either by changing the speed of the pump or the discharge pressure. With reciprocating steam pumps, the pressure is usually approximately constant, and the water load varies with the quantity of water demanded. Either the pressure or the speed may be the independent variable during test, according to the operating conditions of the pump.

The pressure may be varied during the test for different runs, by opening the stop valve in the discharge pipe more or less.

The quantity of water delivered per minute may be measured by any of the methods of determining water rates. For large pumps, weirs may be used.

The head against which the water is discharged may be measured with a pressure gage set in the discharge pipe.* The suction head generally may be measured in feet, and taken as the difference in level between that of the water supplied and the center of the pressure gage used for measuring discharge pressure. Since 1 lb. per square inch = 2.3 feet head, the total head is

$$D = 2.3p + D'$$

in which D' is the suction head in feet.

Readings of p and D' should be taken at regular intervals during each run.

These quantities determine the water horse-power according to the formula previously given. The water horse-power may be estimated, according to the second formula, if the discharge pressure and the number of strokes per minute are measured, and if a value for the percentage of slip is assumed (see (*d*)).

Neither of the formulas for water horse-power takes into account the kinetic energy of the water delivered. Generally the velocities of the water are low enough to make this a negligible quantity. If it is desired to calculate the horse-power due to kinetic energy, the velocity should first be figured from the cubic feet discharge per unit of time and the cross-section of the pipe. The result is then obtained by,

$$\text{W.h.p. available from kinetic energy} = \frac{Wv^2}{2g} \div 33{,}000$$

in which W has the same value as before, v is the velocity in feet per second, and $g = 32.2$, nearly.

* See also Test 70 (*c*).

(b) Mechanical and Fluid Losses. The loss due to mechanical friction of the pump parts may be obtained by indicating both the water and steam ends of the pump. The indicated horse-power of the water end is obtained in exactly the same way as for the steam cylinder, the same values being used for the number of strokes per minute and the average length of stroke. Then the steam end I.h.p. minus the water end I.h.p. equals the mechanical friction horse-power.

The fluid losses are due to leakage, or slip, fluid friction of the water against its passage in the ports, eddies, etc. Slip causes loss of power through water being pumped against pressure from one part of the pump to another without being discharged into the delivery main. These losses are included in the power shown by the indicator diagram for the water end. Consequently the horse-power lost equals the I.h.p. of the water end minus the W.h.p. as calculated under (*a*).

(c) The Gross Efficiency for each value of W.h.p. is obtained by dividing the W.h.p. by the corresponding I.h.p. of the steam end. It is sufficient, for each run, to use average values of heads, water rates, mean effective pressures, and stroke lengths and speeds with which to calculate a single value of efficiency.

The indicated horse-power of the water cylinders is sometimes measured. This divided by the I.h.p. of the steam cylinders is the efficiency covering mechanical losses only.

(d) Slip. The piston displacement in cubic feet is $\frac{LAN}{144}$ per minute. The water actually displaced, in the same units, is $W/62.5$ at ordinary temperatures. The percentage of slip is therefore,

$$\frac{\frac{LAN}{144}-\frac{W}{62.5}}{\frac{LAN}{144}}\times 100=\left(1-\frac{2.3W}{LAN}\right)\times 100.$$

The quantities involved have been obtained for the other determinations.

(e) **The Capacity** of pumps is generally expressed in gallons per twenty-four hours. Knowing the weight of water, or cubic feet, discharged per minute, the capacity is readily calculated, for which purpose the following closely approximate relations may be used.

1 gal. weighs 8.33 lbs. 1 cu.ft. contains 7.5 gal.

Problem 50_1. The following data are obtained from the test of a duplex pump. Discharge pressure, gage, 35 lbs.; suction lift, 4.5 ft.; water discharged per minute, 110 lbs.; length of stroke, 5 ins.; number of strokes per minute, both cylinders, 250; size of water cylinders, 2 ins.$\times$5 ins. What is the W.h.p., the percentage of slip, and the capacity of the pump?

Ans., 0.283 H.p.; 22.6%.

Problem 50_2. In the preceding problem, if the internal diameter of the discharge pipe is 1.38 ins., what is the horse-power due to velocity?

Problem 50_3. If the pressure in the discharge pipe is 12 lbs., and if the water supply is 6 feet above the gage indicating the discharge pressure, what is the total head? *Ans.*, 21.6 ft.

Problem 50_4. Run a series of tests on a steam pump to show the relation between slip and discharge pressure, and slip and speed.

51. * Economy Test of a Steam Pump

Principles. These, in general, are the same as given for the economy test of a steam engine, Test 45, the only difference being that the useful work is measured in terms of water horse-power as defined under Test 50.

In addition to the measurements listed under Test 45, another one is usually required, namely, of "duty." Duty is the number of foot-pounds of useful work performed by a pump per million B.t.u. consumed by the engine.

(a) **Steam and Heat Consumption.** The pounds of steam consumed per hour may be found by any of the methods under Test 45 (*a*). The water horse-power is determined as under

* See also Appendix B, items 21, 39 and 43.

Test 50 (*a*). From these two quantities the pounds of steam per W.h.p.-hour may be calculated.

The heat consumed per pound of steam is $H-h'$, using the same notation and reasoning as given under Test 45, principles. Pressure readings of the supply and exhaust steam, and quality determinations of the supply steam are required for these heat quantities.

(b) **Thermal Efficiency.** This is the same as for Test 45 (*c*) except that the Rankine standard need not be used in most cases.

(c) **Duty.*** The thermal efficiency (as stated above), when expressed as a fraction, is the heat equivalent of the useful work done per B.t.u. consumed. The number of foot-pounds of useful work done by each B.t.u. consumed is therefore 778 times the efficiency, 778 being the mechanical equivalent of heat. Since the duty is the number of foot-pounds per million B.t.u. consumed,

$$\text{Duty} = 778{,}000{,}000 \times \text{Efficiency}.$$

It should be noted that the efficiency is here expressed not as a per cent, but as a fraction of the heat consumed.

Problem 51_1. The steam used for the test cited in Problem 50_1 was 88 lbs. per hour, and was at 85 lbs. gage pressure, quality 97 per cent. The pump exhaust steam was at 10 lbs. gage. What were the steam consumption in pounds per horse-power-hour, the heat consumption in B.t.u. per horse-power-hour, the thermal efficiency, and the duty?

Ans., Eff. = 0.86%.

Problem 51_2. A pump discharges 4750 lbs. of water against a total head of 100 ft., in a certain time. During the same time it is supplied with 100 lbs. of steam and from each pound it consumes 1000 B.t.u. What is the duty of the pump? *Ans.*, 4,750,000.

52. Economy Test of an Injector

Principles. The injector is a pump in which the heat energy of steam is directly used. Works on steam engines and steam

* Item 43(c), Appendix B, is intended to replace "Duty."

boilers describe the instrument in detail. The results from an injector test are the same as those from a steam pump test. In addition, the number of pounds of water pumped per pound of steam supplied is usually quoted. The methods of testing are somewhat different, inasmuch as a direct measurement of the weight of steam per hour is not necessary, this quantity being obtained from the heat balance.

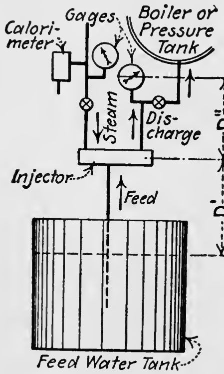


FIG. 99.—Injector Test.

Fig. 99 shows diagrammatically the arrangement of an injector for test.

An expression for the weight of steam used per minute is deduced by equating the heat energy of the steam entering the injector, plus the mechanical and heat energy of the water entering, to the heat and mechanical energy of the water discharged. The datum of the heat measurements is at 32° F., and of the energy measurements is the level of the injector.

Let W = weights, in pounds per minute;
h = heat of the liquid;
L = heat of vaporization of the steam supplied;
x = quality of the steam supplied, if wet;
p = pressure of discharge, pounds per square inch;
D', D'' = suction head, and distance between injector and gage, feet, respectively;
t = temperatures, degrees F.

Subscripts f, s, d refer to feed water, steam, and discharge, respectively.

Then, expressing all energies in heat units, and neglecting kinetic energy

$$W_f h_f - \frac{W_f D'}{778} + W_s(h_s + xL) = W_d h_d + \frac{W_d(2.3p + D'')}{778} + \text{Radiation}.$$

Also,

$$W_d = W_f + W_s.$$

Solving these equations simultaneously and neglecting radiation, we have

$$W_s = \frac{h_d - h_f + \dfrac{2.3p + D''}{778} + \dfrac{D'}{778}}{h_s + xL - h_d - \dfrac{2.3p + D''}{778}} \times W_f.$$

In this expression, the heat equivalents of the work are very small compared with the other quantities and they may be omitted. For the heat of the liquid, h, in each case may be substituted its closely approximate value, $t-32$. There results,

$$W_s = \frac{t_d - t_f}{t_s + xL_d - t} \times W_f.$$

Selection of the Independent Variable. This may be the discharge pressure, the feed water temperature, the suction head, or the steam pressure. When the injector is used as an ordinary pump, the discharge pressure is perhaps the most important of these. If this is selected as the independent variable, it may be controlled by means of the valve in the discharge pipe, see Fig. 99. If the steam pressure is selected, it may be varied by the stop valve in the steam pipe, but the gage should be placed on the other side of the valve to record the pressure on the injector. Whichever quantity is selected, all of the others should be kept as constant as possible throughout each run, and the injector should be regulated before each run so as to secure maximum flow.

(a) Pounds of Water Pumped per Pound of Steam. It must first be decided whether to credit the injector with the feed water only, or with the feed water plus the condensed steam which appears in the discharge. If the injector is used for *emptying*, as when draining a mine, the former amount is logically

used; if for *filling,* as when supplying a tank at a height, the latter amount is proper. When used as a boiler feed, there is room for some discussion of which is the correct amount. The condensed steam is returned to the boiler, but it is used again to operate the injector, so strictly speaking it is not a part of the useful feed water. On the other hand, a duplex pump used for feed water would *not* return the steam used to drive it and would have to pump that much more water.

From the equation previously deduced, we have

$$\frac{W_f}{W_s}=\frac{t_s+xL-t_d}{t_d-t_f},$$

which is the pounds of feed water pumped by one pound of steam. Adding one to the numerical value of this expression gives the pounds of water plus steam pumped.

The temperatures are obtained from thermometers appropriately placed in the discharge, feed, and steam pipe; the latent heat, from the steam tables; and the quality, from a calorimeter. It is often sufficient to assume a value for the quality, provided that may be done within 2 per cent of its true value. The injector should be run long enough before each run at the required conditions to establish uniform temperatures. The run should be long enough to secure enough temperature readings for a fair average, and for the proper measurement of the quantities mentioned later.

(b) Pounds of Steam per Water Horse-power-hour. The equation for W_s, previously deduced, gives the pounds of steam used per minute. The water horse-power is

$$\text{W.h.p.}=\frac{W_f(2.3p+D'+D'')}{33{,}000}\quad\text{or}\quad\frac{W_fD'+W_d(2.3p+D'')}{33{,}000},$$

the one or the other to be used according to the considerations

given under (*a*). The pounds of steam per water horse-power hour is then

$$S=\frac{60W_s}{\text{W.h.p.}}.$$

For this quantity it is necessary to make the same measurements as enumerated under (*a*) and also to measure the feed water rate and the heads, D' and D''. The pounds of feed water per minute may be determined by any of the methods for measuring water. A convenient arrangement is to use a hook gage or water glass in the tank from which the feed is obtained, by which the drop in water level may be timed. This causes a variation in the suction head, but if the tank is of generous cross-section, the percentage of variation may be small enough not to effect the results materially. The average value of the suction head may be figured from the readings of water level. The head D'' is, of course, constant throughout the test.

(c) **Thermal Efficiency.** When considered merely as a pump, the same expression for efficiency as for a steam engine holds, Test 45 (*c*).

$$\text{Efficiency}=\frac{2545}{S(h_s+xL-h_d)}.$$

When considered as a boiler feed apparatus, the efficiency is much higher than this because the heat in the discharge is then "useful," and the only loss is that due to radiation. A determination of radiation may be made from the heat balance if the steam supplied is measured as well as the feed water, from which the boiler feed efficiency may be found. This requires much more precision in all the test measurements than the other results do, however, because the radiation is a small difference between two large quantities.

(d) **Duty**. See Test 51 (*c*).

(e) **Capacity** should be figured in gallons per twenty-four

hours from the test results of W_f or W_d, according to the use to which the injector is to be put.

Problem 52$_1$. A test of an injector yields the following data. Steam pressure 100 lbs. gage; quality, 97 per cent; temperature of feed, 60° F.; temperature of discharge, 147° F.; pressure of discharge, 105 lbs.; gage in discharge pipe, 5 ft. above injector; level of feed water, 10 ft. *above* injector; cross-section of supply tank, 7.1 sq. ft.; rate of fall of water in supply tank while pumping, 4.12 in. per minute. What is the ratio of water pumped to steam used, the pounds of steam per W.h.p.-hour, the efficiency, and the duty? Base all results on the total amount of discharge. *Ans.*, Eff. = 0.38%.

Problem 52$_2$. What should be the capacity of an injector to supply a 100-H.p. boiler, if it operates only two-thirds of the time?

53. Economy Test of the Pulsometer

Principles. The pulsometer is a pump in which the pressure energy of steam is applied directly to the water pumped without intermediate pistons. It consists of two cast-iron chambers controlled by automatic valves and working alternately. Steam, entering one, discharges the water by its superior pressure, while in the other, the partial vacuum, formed from steam condensed by contact with the water, draws a fresh supply of feed.

The results from a pulsometer test, the equations to be used, and the method of testing are exactly the same as for an injector test, 52, to which the reader is referred.

54. * Economy Test of a Steam Boiler

Principles. The heat available in the coal consumed in the furnace of a steam boiler is distributed under four heads: First, the useful heat which goes to evaporate the feed water; second, the heat lost with the carbon slipping through the grate or removed in cleaning; third, the heat carried up the stack by the exhaust gases; and fourth, the heat lost by radiation and all otherwise unaccounted for losses. The heat carried up the stack may be traced under the subheads, heat carried as tem-

* See Appendix B, items 16, 35 and references.

perature of the dry exhaust gases, heat carried by steam formed by evaporation of water in the coal and from the combustion of hydrogen, and heat lost through incomplete combustion of carbon, hydrogen, and hydro-carbons.

A complete boiler test to determine these quantities includes measurements of feed water and coal supplied for a counted time, and analyses of the coal, ash, and flue gases.

Besides the heat balance, certain other quantities are to be found for the purpose of comparing the general performance of the boiler with that of others. The following terms should be understood in this connection.

A unit of evaporation (U.e.) is 970.4 B.t.u. added to the feed water. This unit equals the latent heat of steam at atmospheric pressure. A unit of evaporation will therefore make one pound of dry steam in a boiler operated at atmospheric pressure and supplied with feed water at 212° F., or, more briefly, "from and at 212 degrees."

The factor of evaporation (F.e.) is the number of units of evaporation added to each pound of the feed water. If H is the total heat of 1 lb. of steam generated under the existing conditions of pressure and quality, and h' the heat of the feed water, both counted from 32 degrees, then the heat added per pound is $H-h'$, and

$$\text{Factor of evaporation}^{*}=\frac{H-h'}{970.4}.$$

Equivalent evaporation (E.e.) may be regarded as the number of units of evaporation added to a stated amount of feed water, and is expressed in units per hour, or per square foot of heating surface, or per pound of fuel, etc. Since each pound of feed water has F.e. units of evaporation added,

$$\text{E.e.}^{*}=\text{No. of pounds of feed water}\times\text{F.e.}$$

* The 1922 Code on Definitions and Values defines the "Unit of Evaporation" as 1000 B.t.u. added to the feed water. The term "factor of evaporation" is discarded.

This quantity also equals the number of pounds of dry steam which would be generated by the transfer of the same quantity of heat in a boiler operating "from and at 212 degrees." Equivalent evaporation is generally referred to as a number of pounds of steam under these conditions.

A boiler horse-power (Bo.h.p.) is generated when 34.5 units of evaporation are transferred to the feed water per hour. Knowing the total number of units of evaporation developed during one hour in any boiler trial, the boiler horse-power is

$$\text{Bo.h.p.} = \text{U.e. per hour} \div 34.5.$$

The numerator of this fraction is equivalent evaporation per hour. Therefore

$$\text{Bo.h.p.} = \text{Pounds of feed water per hour} \times \text{F.e.} \div 34.5.$$

If in this expression we put Bo.h.p. = 1, then the pounds of feed water per hour equals $\frac{34.5}{\text{F.e.}}$. That is

The number of pounds of feed water to be evaporated in one hour to make one boiler horse-power equals 34.5 *divided by the factor of evaporation, the latter quantity being fixed by the conditions of feed water and steam.*

Also, the number of B.t.u. to be transferred per hour to make one boiler horse-power = 34.5 × 970.4 = 33480 B.t.u.

The rate of combustion for coal-fired furnaces is the number of pounds of coal consumed per hour per square foot of grate.

The unit of capacity "Boiler horse-power" is also discarded in favor of a rating in "B.t.u. per hour" = weight of feed per hour × (H − h′). Another rating is recommended, as well: "Units of Evaporation per hour" = weight of feed per hour × (H − h′) ÷ 1000.

The 1922 Boiler Test Code (at date of printing) does not entirely conform with the above. Consequently, the terms in present use are given in the text. For the new usage, see Appendix B, item 16 and Paras. 113, 130.

Duration of test is governed by the error of starting and stopping. All the conditions of the boiler and furnace should be the same at the end of the test as they were at the beginning, especially in regard to the quantity and quality of coal on the grate, and water in the boiler. If the coal remaining on the grate at the end of the test is greater than that on the grate at the start, the furnace will be charged with a greater amount than it has actually consumed, and vice versa, unless an estimate is made of the difference between the amounts. The probable error of such an estimate is greater in the case of the coal than of the water. Consequently, the error in the coal measurement determines the duration of the test. The test should be long enough to make the error of starting and stopping less than 1 or 2 per cent of the total amount of coal fired. Assuming this error to correspond to a difference of 1 in. in the fuel bed, the resulting error in the coal quantity will be 2.5 lbs. for each square foot of grate surface, the weight of a cubic foot of incandescent coal being about 30 lbs. Two and one-half pounds is 1 per cent of 250 lbs., so the test should be continued until a total of 250 lbs. of coal per square foot of grate surface has been fired. The higher the rate of combustion, the shorter is the duration. If the rate is 20, for example, the test should be about twelve hours long.

In all cases, the length of the test should be a multiple of the regular cleaning period in order to cover and keep constant regular operating conditions. In the example just cited, if the cleaning period is eight hours, the test should be continued for sixteen hours.

Starting and Stopping. The boiler should be operated during a preliminary run and complete readings taken until they show uniformity of all conditions. The furnace should then be thoroughly cleaned, enough live coal being left on the grate, as in usual cleaning, to start a new fire. The thickness of this coal bed should be estimated and noted, and the level of the water

in the water column marked with a string tied around the glass. The boiler is then fired with a fresh charge of weighed coal, the time of firing the first shovelful of which is taken as the start of the main test. The ash-pit should then be cleaned and all temperatures, pressures, etc., read. At the end of the test, the cleaning is repeated in exactly the same way as at the beginning, the time of firing the first shovelful of the fresh coal after cleaning being taken as the end of the test. The same observations are made as at the beginning. It is best to note the water level just after the cleaning, with the fire door open, because then there will be a minimum of ebullition to cause a false level.

Sampling of the coal should be done as for the proximate analysis Test 37, page 165. Sampling of the flue gas is described under Test 40, page 193. Sampling of the ashes and refuse removed in cleaning and falling through the grate is according to the same principles as for coal. The steam should be sampled for quality determinations, page 144.

To make clear the quantities resulting from a boiler test, a set of calculations from observations made for an actual test will be given for each quantity analyzed in the following. The condensed or average observations are given below.

DATA FROM A BOILER TEST AND NOTATION USED IN FORMULAS

Duration of test, hours	=12
Total weight of coal as fired, pounds	=4925
Total weight ash and refuse, pounds	=734
Total weight water evaporated, pounds	=36,400
Boiler pressure, pounds per square inch, abs.	=110
Quality of steam, x	=0.992
t_f = Temperature of feed water, degrees F.	=98
t = temperature boiler room, degrees F.	=80
T_e = temperature flue gas, degrees F.	=680
Proximate analysis. Percentages by weight of coal as fired.	
Moisture	= 2.75
Volatile matter	= 6.00
Fixed carbon	=78.45
Ash	=12.8

Weight of carbon in 1 lb. of refuse.................... = 0.20
m, *vm*, *fc* = weights of moisture, volatile matter, and fixed carbon per pound of dry coal, respectively.
Heat value of coal by calorimeter.................... = 13,040

Volume Analysis of the Flue Gas, Per Cents

D = carbon dioxide.................................... = 11.0
O = oxygen.................................... = 9.0
M = carbon monoxide.................................... = 0.5
N = nitrogen.................................... = 79.5

Weights, in Pounds, of Carbon from 1 lb. of Dry Coal

C_a = carbon wasted in ash;
C_t = total carbon; fixed, and in volatile matter;
C_g = carbon burned, appearing in flue gas.

(a) **Determination of Total Coal and Refuse.** The coal to be fired is weighed out by the barrowful as it is needed and heaped on the boiler room floor at a place convenient for firing, one barrowful at a time. The following is the best form of log for coal records. It includes a few observations to illustrate its purpose.

Weight of Barrow.		Net Weight.	Total Weight.	Time of Firing First Shovelful.
Empty.	Full.			
Test started..	....	....	....	8:00 A.M.
100	250	150	150	8:00
100	275	175	325	8:24
100	260	160	485	8:46
Etc.				

A chart should be plotted as the test proceeds of total coal as shown by the fourth column against time as a base. It should be noted particularly that the time to be plotted against the total weight from each barrowful is not the time of firing the first

shovelful from the corresponding weighing of coal, because this coal is not entirely burned until the first shovelful of the next charge is fired. Referring to the log, 150 lbs. should therefore be plotted against 8:24; 325 lbs. against 8:46; and so forth. If the rate of evaporation and all other conditions external to the furnace are uniform, then with proper firing, the rate of combustion will be uniform, and a fair straight line will be represented by the points of the plot. This line should pass through the origin; if it intersects the axis of coal weights above the time axis, it indicates that there was too much coal on the grates at the start; if below, too little. In such a case, it is better to figure the total coal from the slant of the line.

The ashes accumulating under the grates and the refuse removed during cleanings in the main test should be collected, allowed to cool without wetting, and then weighed.

(b) Determination of Total Feed Water Evaporated. The feed water may be measured by a meter in the feed water line, but when certainty of accurate results is desired, a direct weighing system such as represented diagrammatically by Fig. **100** should be used. With this arrangement, the levels of the water at the start of the test is marked by the gage, *g*, in the suction tank and by the string on the water column at *m*. The feed pump should be operated at a constant rate through the test, this rate being determined by the horse-power required of the boiler; and, with proper firing and regulation, the water level shown by the column should always be approximately at *m*. The best form of log for the records is as follows. A few sample observations are included to illustrate.

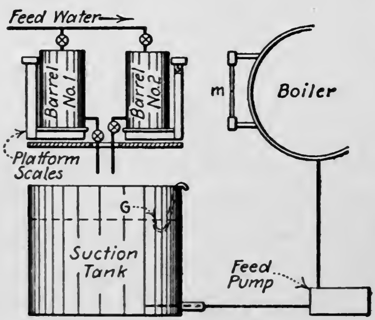


Fig. 100.—Weighing System.

Barrel No.	Weight of Barrel.		Net Weight.	Total Weight.	Time of Passing Gage Level.
	Empty. Lbs.	Full. Lbs.			
Test started	....	....	....	....	8:00 A.M.
1	50	300	250	250	8:12
2	60	320	260	510	8:25
1	75	300	225	735	8:34
Etc.					

As the test proceeds, a chart should be plotted of total water as shown by the fifth column against the time taken to evaporate it as a base. When the first charge, 250 lbs. on the log, is turned into the suction tank, the level is raised above the gage, *g*, Fig. 100. When the level has again descended to the gage, all of the 250 lbs. has then been evaporated provided that none of it has accumulated in the boiler as would be shown by a higher level in the column than *m*. The time of passing the gage level in the suction tank is therefore the time to be plotted; 250 lbs. against 8:12; 510 lbs. against 8:25; and so forth.

It is especially important to keep the rate of evaporation uniform for a valid test. The water-time curve should therefore be a straight line.

If the water level is not the same at the end as at the beginning of the test, the total water may be figured from the slant of the line, or a correction may be applied to the last figure in the fifth column of the log. The correction is calculated from the cubical contents of the boiler between the two levels and the density of the water at the existing temperature. It should be noted that a false level may appear in the water column if the ebullition is violent or if the water column is blown down within a short time before reading. The latter occurrence is due to the fact that blowing down forces into the glass hotter and less dense water than is there normally.

(c) **Quantities to be Found Prior to Calculations of Results.** The proximate analysis of the coal is generally expressed in percentage by weight of the coal "as fired," that is, including moisture. To find the weights per pound of dry coal, it is necessary only to divide each of the items by 100 minus the percentage of moisture. Thus, for the data of the test previously cited,

$$\begin{aligned} m &= 2.75-(100-2.75)=0.0283 \text{ lb. of moisture;} \\ vm &= 0.0617 \text{ lb. of volatile matter;} \\ fc &= 0.807 \text{ lb. of fixed carbon;} \\ a &= 0.132 \text{ lb. of ash;} \\ vm+fc &= 0.869 \text{ lb. of combustible.} \end{aligned}$$

It is necessary to know the **total carbon per pound of dry coal.** If the ultimate analysis has been made, this figure is available; otherwise, it is necessary to estimate it. The method quoted under Test 37 (*b*) may be used, if the proximate analysis only has been made.

On page 169, the data of the above analysis are used as an example of this method, and it was found that the total weight of carbon in 1 lb. of dry coal is

$$C_t = 0.807+0.01\times 0.869 = 0.816 \text{ lb.}$$

The proportion of **carbon** which is **wasted through cleaning and grate,** C_a, may be found in two ways. The first method is as follows.

The total carbon in the refuse equals the total amount of refuse minus the ash from the total coal fired. The ash from the total coal fired equals the total quantity of coal times the proportion of ash in it as given by the proximate analysis. Then,

$$C_a = \frac{\text{Total refuse, lbs.} - \text{Total coal, lbs.} \times \text{Per cent ash} \div 100}{\text{Total coal, lbs.} \times (1 - \text{Per cent moisture in coal} \div 100)},$$

the denominator being the total weight of dry coal.

The second method uses the proximate analysis of the ash.

$$C_a=\frac{\text{Total refuse, lbs.}\times\text{Weight of C in 1 lb. refuse}}{\text{Total coal, lbs.}\times(1-\text{Per cent moisture in coal}\div 100)}.$$

In the first equation it is assumed that all of the ash coming from the coal fired during the main test is removed with the refuse. This, however, is uncertain, so it is better to use the second method. Applying the data of the test under consideration, each pound of the refuse was found to contain 0.20 lb. of carbon. The total coal as fired was 4925 lbs. and it contained 2.75 per cent of moisture. The total weight of refuse was 734 lbs. Then

$$C_a=\frac{734\times 0.20}{4925-.0275\times 4925}=.0307 \text{ lb.}$$

To find the **weight of the carbon burned.**

$$C_g=C_t-C_a.$$

For given data $C_g=0.816-0.0307=0.785.$

To calculate the factor of evaporation, the heat content of the steam generated is first found. Referring to the test data, page 266, the steam tables give $H=305.5+0.992\times 882.5=1181.$ Consequently

$$\text{F.e.}=\frac{1181-(98-32)}{970.4}=1.15.$$

All the items of the heat balance will be figured on the basis of 1 lb. of dry coal. The heat value of the dry coal as found by calorimeter or calculation (see Test 38) will therefore be the heat supplied.

(d) Equivalent Evaporation, per Pound of Coal as Fired, is

$$\text{E.e.c.}=\frac{\text{Total water evaporated, lbs.}\times\text{F.e.}}{\text{Total coal as fired, lbs.}}.$$

The units of this expression are either pounds of water evaporated from and at 212 degrees per pound of coal, or units of evaporation per pound of coal. For the given data (page 266),

$$\text{E.e.c.}=\frac{36400\times 1.15}{4925}=8.5 \text{ lbs. or U.e.}$$

Per pound of dry coal

$$\text{E.e.d.c.}=\text{E.e.c.}\div(1-\text{moisture, lbs.}),$$

For the given data, E.e.d.c. $=8.5\div(1-.0275)=8.65$ lbs. or U.e.

Per pound of combustible

$$\text{E.e.cb.}=\text{E.e.d.c.}\div(vm+fc),$$

For the given data, E.e.cb. $=8.65\div 0.869=9.95$ lbs. or U.e. per hr.

Per hour

$$\text{E.e.hr.}=\frac{\text{Total water evaporated, lbs.}\times\text{F.e.}}{\text{Total time, hours.}}.$$

For the given data,

$$\text{E.e.hr.}=(36400\times 1.15)\div 12=3440 \text{ lbs. or U.e. per hr.}$$

(e) **Over-all Efficiency.** Since there are E.e.d.c.$\times$970.4 B.t.u. added to the feed water for each pound of dry coal fired, and since the heat available from this coal is its heat value, the efficiency is,

$$\frac{\text{E.e.d.c.}\times 970.4}{\text{Heat value of dry coal}}.$$

For the given data, over-all efficiency $=8.65\times 970.4\div 13{,}040=65$ per cent.

(f) **Loss to Carbon Wasted in Refuse.** Taking the heat value of carbon as 14,600 B.t.u., this loss per pound of dry coal is

$$14{,}600C_a.$$

For the given data, $14{,}600 \times 0.0307 = 448$ B.t.u., or $448 \div 13{,}040 = 3.4$ per cent.

(g) Loss of Heat to Dry Exhaust Gases. This determination involves the measurement of the excess of temperature of the flue gas over that of the boiler room, and of the weight of dry flue gas per pound of dry coal, W_d. For the specific heat, an average value, 0.24, may be used with sufficient accuracy. Then the loss is

$$0.24 \times W_d \times (T_e - t).$$

To find W_d the formula given on page 199 may be used. For the given data

$$W_d = \frac{3.04 \times 79.5 \times .785}{11.0 + 0.5} + .785 = 17.3 \text{ lbs. per lb. of dry coal}$$

and the loss is $0.24 \times 17.3 \times (680 - 80) = 2490$ B.t.u. or $2490 \div 13{,}040 = 19.1$ per cent.

(h) Loss of Heat to Water Vapor in the Flue Gases. To calculate this, we must know the weight of vapor per pound of dry coal and its heat content above room temperature, including latent heat. The vapor carried in as humidity of the air need not be considered because it is small in quantity and its latent heat is not added by the coal. This leaves only the vapor from the moisture in the coal and that due to combustion of hydrogen to consider. From the formulas for W_v, the weight of vapor per pound of coal, on page 200, and for the heat of steam when mixed with flue gases on page 143, we have

$$\text{Loss per pound dry coal} = (9H_t + m)(1090 + 0.46T_e - t).$$

H_t, the weight of hydrogen per pound of dry coal, may be determined according to Test 37 (b) in the case of semi-anthracite and bituminous coals if the ultimate analysis has not been made. For the proximate analysis of the given test, the calculation for H_t is made on p. 169, resulting in $H_t = 0.025$. Therefore the heat

lost is $(9\times0.025+0.0283)\ (1090+0.46\times680-80)=335$ B.t.u. or $335\div13{,}040=2.6$ per cent.

(i) **Loss of Heat Due to Incomplete Combustion.** Considering only the incomplete combustion of carbon, the loss of heat when 1 lb. of carbon burns to CO instead of CO_2 is $14{,}600-4400=10{,}200$ B.t.u. (see page 169). In the boiler test there are W_8 lbs. of carbon per pound of dry coal burned to CO. An expression for W_8 is derived on page 200. The loss is

$$10{,}200\times W_8 \text{ B.t.u. per pound of dry coal.}$$

For the given data,

$$\text{Heat lost}=10{,}200\times\frac{0.5\times0.785}{11+0.5}=338 \text{ B.t.u. or } 338\div13{,}040$$

$=2.7$ per cent.

(j) **Radiation** is found by difference. For the given data, it is

$$100\%-65\%-3.4\%-19.1\%-2.6\%-2.7\%=7.2\%$$

(k) **Other Quantities. The boiler horse-power** is, as previously defined,

$$\frac{3440}{34.5}=99.9 \text{ B.h.p.}$$

The equivalent evaporation per hour per square foot of heating surface is,

$$\frac{\text{E.e.hr.}}{\text{Sq.ft. of surface}}.$$

For the given boiler, the heating surface was 1000 sq. ft., so E.e.hr. per square foot $=3440\div1000=3.44$ lbs. or U.e.

The grate surface of this boiler was 30 sq. ft., so the **rate of combustion** of dry coal, as previously defined, was $399\div30$ $=13.3$ lbs. per hour per square foot.

Efficiency of the Grate and Cleaning. The proportion of the combustible lost is $C_a \div (vm+fc)$. Consequently this efficiency is,

$$1-\frac{C_a}{vm+fc}$$

For the given data, this is $1-0.0307 \div 0.869 = 96.5$ per cent.

Efficiency of the boiler and furnace (exclusive of grate) is the over-all efficiency divided by the efficiency of the grate. For the given data, this is $65 \div 96.5 = 67.4$ per cent.

This efficiency is often defined as the heat absorbed per pound of the combustible burned divided by the heat value per pound of combustible.

The excess coefficient may be found from the expression derived on page 201.

Besides the total water and coal curves, it is well to plot, after the test, all the readings of pressure, feed water, flue and boiler room temperatures, per cent CO_2 and the draft in inches of water. These will be broken curves. They will aid deductions of causes of any irregularities in the performance.

A few barometer readings are desirable in connection with natural draft.

All circumstances that may affect the conditions of the test should be carefully noted for possible future reference.

Problem 54_1. A 50-H.p. boiler delivers steam at 175 lbs. gage, superheated 100°, from feed water at 50°. What is the F.e.? How many pounds of feed water will be needed per hour during a test? Assuming an efficiency of 70 per cent and anthracite coal with 15 per cent ash, about how many pounds of coal will be needed per hour? *Ans.*, 1.27; 1360 and 190 lbs.

Problem 54_2. For a given natural draft, there is a corresponding flue temperature necessary to maintain it. If the excess air during the operation of a boiler varies, and all other conditions remain constant, deduce the relation between per cent CO_2 and efficiency. Note that efficiency $=1-$ a constant $-$ loss to dry exhaust gases, approximately.

55. * Test of a Surface Condenser.

Principles. A steam condenser receives, in practice, a mixture of steam and water and air. The steam entering the condenser, if exhausted from a steam engine or turbine, carries with it moisture. Air enters the system through leaks in steam-pipe joints, stuffing boxes, and from the boiler feed.

Since it is the function of a condenser (considered as a power unit) to maintain a vacuum, this is one of the most important items to investigate. Under ideal conditions of heat transfer, the vacuum possible to be maintained would be that of saturated steam corresponding to the temperature of the outgoing condenser water. This ideal is not realized for two reasons. First, a *difference* of temperature between the steam and the cooling water is required that there should be a heat flow from the one to the other. Second, the effect of air mixed with the steam is to make a higher pressure than that due to the steam alone, according to the law of partial pressures (see p. 142). If, for example, the temperature in the steam space of a condenser were 126° F. the corresponding steam pressure (from the tables, p. 370) is 2 lbs. absolute, very nearly. Suppose there is air present, in amount equal to one-quarter by weight, of the steam. A cubic foot of the steam space would then contain a weight of air equal to one-quarter of the density of steam at 2 lbs., absolute, and the specific volume of the air would be four times that of the steam, that is, $4 \times 174 = 696$ cu.ft. per pound. Consequently (from $PV = RT$).

$$\text{Pressure of the air} = \frac{53.4 \times (126+460)}{144 \times 696} = .31 \text{ lb. per square inch}$$

and the pressure of steam and air combined would be $2 + .31 = 2.31$ as compared with 2 lbs. if the steam space contained steam alone. It follows, then, that the effect of air upon condenser performance is to increase the absolute pressure above that corresponding to

* See also Appendix B, items 28 and 52.

the prevailing temperature, and therefore to lessen the effectiveness of the condenser.

The question of the most economical vacuum is a pertinent one in connection with condenser tests. This leads to a consideration of the cost of cooling water, and of auxiliary power, that is, of circulating, wet air, and dry-air pumps.

Finally, the rate of heat transmission should be investigated in order to determine the effectiveness of the cooling surface.

The Independent Variable may be the vacuum, weight of condensate, or the quantity of cooling water per unit of time.

The following notation will be used in this and the next test.

W_w = weight of cooling water, pounds per hour;
W_s = weight of wet steam entering, pounds per hour;
W_r = weight of cooling water per pound of condensate;
T_s = temperature of the steam in the condenser, degrees F.;
t_s = temperature of the condensate discharged;
T_i = temperature of cooling water at inlet, degrees F.;
T_o = temperature of cooling water at outlet, degrees F.;
T_m = mean temperature difference between steam space and cooling water, degrees F.;
x, L = the quality and latent heat of the entering steam;
B = heat transmitted, B.t.u. per hour;
A = area cooling surface, square feet.

(a) Ideal and Actual Vacuums. The actual vacuum may be measured with a calibrated gage, its indications being reduced to absolute pressure by subtracting them from the barometric pressure as observed. Or, better yet, an "absolute pressure gage" may be used. This may be in the form of a glass U-tube with one leg sealed at the top and completely filled with mercury. A sufficient reduction of pressure on the open leg will lower the mercury column in the filled one; the difference of level in the two legs of the U-tube then is the absolute pressure. A heavy rubber tube, capable of withstanding 15 lb. collapsing pressure, may be

used to connect the open leg of the U-tube to the condenser steam space. The steam inlet pipe should be tapped, close to the condenser, for a $\frac{1}{8}$-inch pipe nipple. Another opening should be made for the insertion of a thermometer.

The condenser should now be operated, as in regular service, at a predetermined vacuum, for a period of say one-half hour. During this time, at intervals of five minutes, temperature readings should be taken of the steam and of the discharge from the wet air pump or hot well pump, the vacuum being maintained constant. If there is but little air in the steam, and the pressure drop through the condenser is small, the absolute pressure as shown by the gage should be but slightly greater than that of saturated steam corresponding to the average temperature of the steam space or of the discharged condensate. A large quantity of air leakage, on the other hand, will be indicated by a material difference between these pressure values.

A series of such tests may be run at various values of the vacuum. It is useful to plot the resulting data, actual against ideal vacuum (corresponding to both T_s and t_s), for the purpose of later comparisons.

(b) Rate of Heat Transmission. This may be found by estimating the heat content of the entering steam (a method is indicated on page 379), or it may be determined by finding the heat added to the cooling water. The water quantities are usually large, and therefore some form of velocity meter is appropriate for measuring the water supplied. Inlet and outlet temperatures of the cooling water (close to the condenser) must be read and averaged. From these data (see notation, page 277) may be calculated the heat transferred per hour.

$$B = W_w(T_v - T_i).$$

(c) Effectiveness of Heat Transmission. This can be judged after determining the number of B.t.u. transmitted per square

foot of surface per hour per degree difference in temperature. This result represents the conductivity of the heat path, and will be referred to as C. Then

$$C=\frac{B}{A\times T_m}.$$

The mean temperature difference, T_m, is found from the relation

$$T_m=\frac{T_o-T_i}{\log_e \dfrac{T_s-T_i}{T_s-T_o}},$$

which is deduced in treatises on heat transmission. Substituting this value and the one previously quoted for B in the equation for C, we have

$$C=\frac{W_w}{A}\times\log_e \frac{T_s-T_i}{T_s-T_o}.$$

The value of C resulting from this equation should be about 300 B.t.u. for ordinary installations, and, in the best designs, as high as 900 B.t.u. Low values indicate fouled surfaces, air pocketing, flooding, etc.

It may be noted from an examination of the equation for C that the more nearly equal are T_s and T_o, the more effective is the heat transmission. These temperatures, by themselves, are therefore a criterion of performance.

Another statement of the effectiveness of heat transmission is in terms of weight of condensate per square foot per hour. This is not very definite, since it depends upon the latent heat of the steam and consequently the vacuum, as well as the quality of the steam. Ordinarily it is about 10 lbs.

(d) **Weight of Condensing Water per Pound of Condensate.** Under ideal conditions, with dry steam, this would be

$$w_r'=\frac{L}{T_s-T_i}.$$

Actually it is

$$w_r = \frac{xL + (T_s - t_s)}{T_o - T_i}$$

plus an amount necessary for cooling the entrained air, radiation, etc. The value of w_r may be calculated from this equation, provided an estimate of x is made (see Ex. 5, page 379), and compared with the ratio of weights of cooling water to condensate as obtained by metering both.

(e) **Leakage Tests** of surface condensers may be made according to the directions on page 231.

Problem 55₁. The gage on a condenser shows 3.45 ins., absolute. The temperature of the steam space is 115°. What percentage of air is present (based on the mixture)? *Ans.*, 20%.

Problem 55₂. A condenser operating at 26 ins. vacuum (normal barometer) takes cooling water at 60° and discharges it at 116°. What is the mean temperature difference? *Ans.*, 29.7°.

Problem 55₃. If in the preceding, 200,000 lbs. of cooling water are used per hour, and the area of the cooling surface is 1000 sq.ft., what is the value of C? *Ans.*, 377 B.t.u.

Problem 55₄. Using data of Problem 55₂, how many pounds of cooling water are required per pound of condensate, assuming dry steam and allowing 12% for heating air, radiation, etc.? *Ans.*, 20.5 lbs.

56. * Test of a Jet Condenser

Principles. These are, in the main, the same as for the preceding, Test 55. There is this difference, however, that, since condensate and cooling water are mixed, the outlet temperature of the latter, T_o, equals that of the former, t_s.

(a) **Ideal and Actual Vacuums.** See Test 55 (*a*).

(b) **Rate of Heat Transmission.** The method of Test 55 (*b*) may be used with the modified formula,

$$B = W_w(t_s - T_i).$$

If the weight of the discharge from the condenser is measured instead of the inlet water, W_w, then steam entering the condenser

* See also Appendix B, items 28 and 52.

must be separately determined and subtracted from the total weight to find W_w.

(c) **Weight of Condensing Water per Pound Condensate.** See Test 55 (*d*). In the formulas quoted t_s may be substituted for T_o since the two are the same.

Problem 56₁. A jet condenser operates under a vacuum of 26 ins. and takes condensing water at 70° F. Assuming the steam dry, and containing no air, how much water is required per pound condensate under ideal conditions?

Ans., 18.2 lbs.

Problem 56₂. A jet condenser operates under a vacuum of 26 ins. and takes condenser water at 70° F. The temperature of the wet air pump discharge is 10° below that of the steam. Allowing 10 per cent for cooling the air, etc., how much water is required per pound of condensate fromdry steam? (Compare with 56₁). *Ans.*, 24.7 lbs.

57. * Test of a Feed-water Heater

Principles. As far as heat transmission is concerned, a closed feed-water heater is identical to a surface condenser; and an open heater to a jet condenser. Tests 55 and 56 should therefore be read in this connection.

The closed feed-water heater generally operates with the water space under pressure, but the open heater runs practically at atmosphere. The steam supplied may be engine or auxiliary exhaust. There may be more steam available than can be used for preheating the water, in which case the excess is either vented to the atmosphere or used in some other apparatus, such as radiators for room heating.

There are so many different combinations for the employment of feed-water heaters that it is impracticable to state a general method of testing covering them all.

The notation on page 277 will be used, in which, for " cooling water " should be understood " feed water," and for " condenser " " feed-water heater."

* See also Appendix B, items 30 and 54.

(a) **Useful Heat Transmitted.** For a closed heater this is, in B.t.u. per hour.

$$B = W_w(T_o - T_i).$$

For an open heater, and for a closed one in which the condensate is returned to the boiler, there should be added to this the heat regained in the condensate.

The quantities in the equation just quoted are to be measured as under Test 55 (*b*).

(b) **Effectiveness of Heat Transmission.** For a closed heater Test 55 (*c*) applies. In either type, the more closely does the outlet temperature of the water approach that of the steam, the more effective is the heat transmission.

(c) **Available Heat.** It is useful, sometimes, to estimate the heat available in the total steam supplied the heater (above feed-water temperature) in order to form an idea of the proportion of it that is utilized. (See Ex. 5, page 377.)

58. Testing the Adjustment of an Internal Combustion Engine

Principles. Most of the modern internal combustion engines work on the Otto cycle which takes four strokes for a complete series of events. First, the suction stroke, in which the piston, advancing from the head end, takes in a charge of gas or vapor fuel mixed with air. Second, the compression stroke, returning, in which the charge is compressed. Third, the expansion stroke, at the beginning of which the mixture is burned and afterward performs the useful work. Fourth, the exhaust stroke, during which the burned gases are discharged. Fig. 101 shows this cycle of operation on a PV diagram. There are two valves to control the charge, one for inlet and one for exhaust. Fig. 102 shows where these valves open and close relative to the crank position and on the indicator diagram. It is seen that the inlet

valve opens a little after dead center. This is arranged so that it will open after the exhaust valve has closed; otherwise there would be a tendency for the exhaust gases to enter the intake. The closing of the inlet valve takes place after crank end dead center so as to provide a good opening for the charge. The corresponding crank angle may be as much as 20° to 30°, with high-speed engines; the momentum of the incoming gases causes them to continue to flow into the cylinder during the inward piston motion. The exhaust valve should open a little before crank end

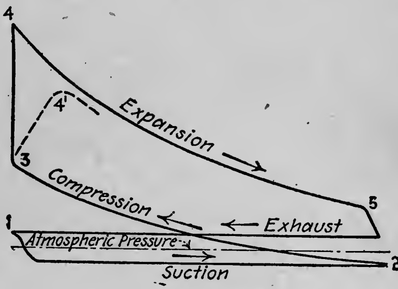


Fig. 101.—Otto Cycle.

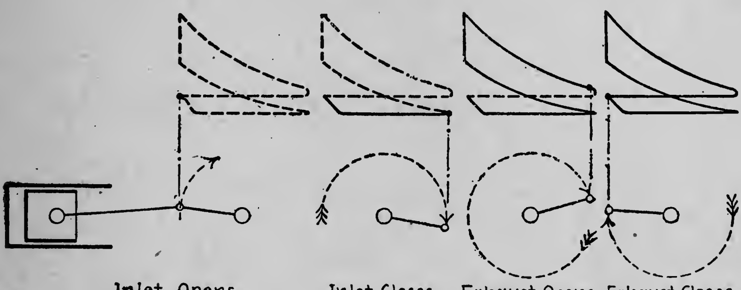


Fig. 102.—Gas Engine Valve Events.

dead center and close a little after the opposite dead center to allow free egress of the burned gases, the amount depending upon the speed and type of the engine. The correct angular distances of the crank from the dead center positions, corresponding to the valve events, are usually stated by the engine manufacturer.

Ignition of the charge should occur in the neighborhood of dead center; generally before, when the engine is running. This

is because it takes an appreciable time for the gas mixture to rise to its maximum pressure after ignition. For maximum power, the greatest pressure should occur at the beginning of the stroke; consequently, ignition should "advance" this position. The greater the rotative speed, the greater will be the angular distance, or advance, between the crank and dead center when ignition takes place. Some fuels burn more slowly than others; for such the ignition must be more advanced. The angular distance varies between zero and 50°.

(a) **Timing the Valves.** This is done by reference to the crank positions when the valves are opening or closing. A prick point should be made on the flywheel rim to mark the line of the crank, and the position of this prick point located by trammels or otherwise when the engine is on dead center. The angular distance between the prick point and its dead center position will determine the motion of the crank for any valve event.

The valves usually are operated by cams on a cam shaft driven by 2 to 1 gears, *a* and *b*, from the main shaft. See Fig. 103. There is a certain amount of clearance between the valve stem and the cam when the latter is not acting. By changing this clearance the timing of the valve will be altered. From the figure it will be seen that to make the clearance greater makes the valve open later and close earlier. The timing may be changed also by changing the relative positions of the gears *a* and *b*. Thus, in the figure, if gear *a* is turned back (clockwise) so that it meshes with one tooth behind on *b*, both opening and closing will be later.

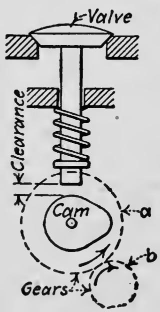


Fig. 103.
Gas Engine Valve.

There is usually some provision made for changing the clearance. The procedure in timing the valves, then, is first to locate the crank positions corresponding to opening and closing, and then

to correct, if necessary, by changing the clearance or gear mesh or both. The point of opening or closing may be accurately fixed by turning the valve stem with the fingers; the friction of the valve when seated is easily felt. It should be noted that the valve events are somewhat different when running because of the expansion of the valve stems due to the working temperature. To allow for this, the engine may be operated until it warms up, and then timed. The makers' instructions for timing are usually for the cold condition.

With high-speed engines, such as are used in automobile and marine practice, it is customary to set the valves by making each clearance a stated amount, and this amount is usually very small, between 0.002 and 0.006 in. The purpose is to minimize the noise and wear which are produced by excessive clearance. Clearances are often made on such engines as small as the mechanic can make them. Systematic procedure calls for the use of "spacer" gages, or "feelers," by the use of which the clearances may be made exact. After such adjustment, the valves should be timed. It is more important to get good timing on exhaust than on inlet valves.

(b) Timing the Ignition. For jump spark ignition with batteries or low-tension magneto, this is readily done by removing the spark plug and noting the crank position to correspond with the occurrence of the spark. For high-tension magneto systems, the timer on the magneto must be observed. For make-and-break systems, it is easy to note the crank position when the igniter breaks contact, as it is visible.

For economical running, ignition should be advanced as far as possible without causing pounding or diminution in speed.

(c) Adjustment of Mixture. Whether the engine uses oil vapor or fuel gas, there is one device for setting the amount of air supplied and another, the amount of fuel. Strictly speaking these should be readjusted for every load on the engine, but usually a single adjustment is made to give the best mixture at

usual running loads. As a general rule, it may be accepted that the most economical mixture is as dilute a one as may be had without causing misfiring or objectionable slowing down of the engine. This effect is due to the lower working temperatures of the expanding medium. Also, for economy, the fuel valve should be open as little as possible. This is especially the case with engines with throttling governors. If the engine is given too much fuel, it tends to speed up which puts upon the governor the work of throttling it so that the suction in the cylinder is greater and the line 1–2, Fig. 1C1, is lower. Thus the greater the throttling, the larger will be the lower loop of the indicator diagram. This represents the work lost in pumping the charge into the cylinder.

With an engine regulated by a governor, the procedure is as follows. The desired load is applied by brake or otherwise and the valve controlling the flow of fuel is set with a slightly larger opening than is thought to be correct. The air opening is then increased until the engine shows signs of slowing down, as determined by a tachometer. Records of speed and valve settings for this trial are kept. The procedure is then repeated with a slightly less fuel valve opening. The smallest fuel valve opening with the largest air valve opening which will not cause objectionable slowing down or irregular firing may be accepted as the best setting for the applied load. In this connection it is essential that the exhaust be observed during each trial; it should be clear and regular.

(d) **Adjustment of Carburetors** for variable speed engines supplied with gasoline, such as automobile and marine engines, is more difficult than the adjustment of mixing valves on constant speed engines, but the same principles are to be observed. Many carburetors are made with automatic adjustment to accommodate changes of speed, and the adjustment is made with the engine delivering no torque. This procedure is convenient, but not productive of best results. In general, it is a good plan to follow the manufacturer's instructions for carburetor adjustment to

get what may be considered an **approximate** setting. A readjustment then should be made, *under a load* representing the average running condition. This readjustment involves reducing the needle valve opening and increasing the auxiliary air opening to a point just short of back firing. This will give the most *economical,* although not the most *powerful,* mixture. Also, it will probably be necessary to determine another setting for starting the engine when cold.

Finally, the action of the carburetor should be judged by the appearance and sound of the exhaust.

(e) Test of Timing by Indicator. This applies to engines running at not more then 250 R.p.m. if the ordinary indicator is to be used. Special gas engine indicators are made to handle about double this speed. Higher than this, only optical indicators may be used with success. A light spring should be used in the indicator, about 20 lbs., and the indicator piston arranged with a stop so that it will not register much above atmospheric pressure, in order to get the suction diagram on a larger scale.

If the inlet valve opens too late, the line from 1, Fig. 101, will not have as sharp a drop as it should and the suction line will be lower. If the inlet valve closes too late, the compression curve will not start sharply from the point 2. If the exhaust valve opens too late, expansion is carried a little beyond point 5, but the beginning of the exhaust line will be higher on account of the greater back pressure. If it closes too late, the effect is much the same as for too late opening of the inlet. When the valve events are too early, opposite effects will be recorded.

There is an important connection between the timing of the exhaust valve and economy in the use of fuel. This has to do with the " scavenging " of the cylinders, that is, the cleaning out of the exhaust gases, which is necessary to the complete combustion of the next charge. Hence the exhaust valve should be kept open for as long a crank motion as other conditions permit.

The indicator diagram does not show fine differences in the

timing of the valves because the events occur near the ends of the stroke where the indicator drum has but little motion corresponding to the angular motion of the crank. For this reason, it is useful to set the reducing motion of the indicator 90 degrees ahead of the crank, so that the valve events will be shown in the middle of the indicator diagram.

If the ignition is properly advanced, the line 3–4, Fig. 101, will be vertical. If ignition is too late, this line will slant as shown by the dotted line 3–4′, causing a diminution of power; if too early it will slant in the opposite direction with the same effect, accompanied by pounding.

Problem 58_1. An inlet valve is found to open too early and close too late. The exhaust valve on the same cylinder opens and closes too early. What should be done to correct the timing, if the two cams are integral with the cam shaft?

Problem 58_2. Draw two indicator diagrams, each superposed on a normal diagram like Fig. 101, to show the effects of too early and too late valve events.

Problem 58_3. Draw a suction diagram to be expected with normal setting when the reducing motion is set 90° ahead of the crank. Draw the work diagram.

59. Mechanical Efficiency Test of an Internal Combustion Engine

Principles. Exactly the same methods and formulas apply to this test as to Test 44, with the exception of the determination of N in the formula for indicated horse-power. Internal combustion engines are generally single acting in moderate sizes, so, with throttling governors and regular firing, there is only one power stroke for each two revolutions of a crank. Under these conditions, the value of N is one-half the revolutions per minute. When the engine governs on the hit-and-miss principle, however, the actual number of explosions per minute must be counted. This also applies with a throttling engine if the firing is irregular.

The indicator diagram, Fig. 101, contains a negative area which must be subtracted from the upper area to get the net mean effective pressure. This may be done with the planimeter by traversing the upper area in a clockwise direction, and the lower, anti-clockwise. The indicated horse-power obtained from the result is the "net" I.h.p. which is greater than the brake horse-power by the horse-power given to mechanical friction. The lower loop of the diagram gives the work lost to fluid friction, that is, to pumping the charge into the cylinder. It may be determined accurately only by using a light indicator spring as explained under Test 58 (*e*).

(a) Determination of Indicated Horse-power. See Test 44 (*a*). If the engine governs on the hit-and-miss principle, the explosions may be counted by timing the sound of the exhaust if the speed is not too fast. Special devices are made for this purpose when a continuous record is wanted. One may be made by tapping a small pipe into the exhaust and covering it with a flap held down by a spring. An explosion causes the flap to rise and so actuate the lever of a revolution counter.

(b) Friction Horse-power. The mechanical friction is obtained as for Test 44 (*b*). The fluid friction may be expressed as horse-power calculated from the mean-effective pressure of the lower loop, the value of N being the same as for the net I.h.p.

(c) Efficiency. (d) Speed Regulation. See Test 44 (*c*) and (*d*).

Problem 59_1. A 2-cylinder, 6-in.×8-in.×350 R.p.m. gas engine, governing on the hit-and-miss principle, misses one out of every five explosion strokes. If the net mean-effective pressure is 20 and 21 lbs. in the two cylinders, what is the I.h.p.? *Ans.*, 3.28 H.p.

Problem 59_2. What is the engine constant of a 6-in.×8-in. gas engine? If the brake horse-power is 5, and the M.e.p. of the upper loop of the diagram is 70, and of the lower loop 5 lbs., and the engine fires 163 times per minute, what is the horse-power lost to mechanical friction? To fluid friction? *Ans.*, .000571, 1 H.p.. .077 H.p.

Problem 59_3. Compare by actual test the speed regulation of an engine working on the hit-and-miss principle with a throttling engine. Determine coefficient of regulation at full and half loads.

60. * Economy Test of a Gas Engine

Principles. A complete economy test of a gas engine should include, besides the fuel consumption, the determination of the various losses so that their distribution may be studied. All of the heat energy of the fuel supplied must be accounted for in the form of useful work and losses. The equation between these quantities is called the *heat balance.*

The losses involved in the operation of a gas engine are as follows. Mechanical friction, heat carried away by jacket water, by the dry exhaust gases, by steam in the exhaust, by unburned fuel gas, and stray losses as radiation, etc. It is convenient and logical to base the calculation of these losses upon 1 cu. ft. of fuel gas under standard conditions of temperature and pressure. The heat balance may then be written, "heat supplied by 1 cu. ft. of gas equals the heat from 1 cu. ft. turned into useful work plus heat from 1 cu. ft. lost to friction plus, etc."

To measure the useful work and friction and jacket losses on this basis it is necessary to meter the fuel supplied in a given time. The other quantities of the heat balance require the analyses of both the fuel and exhaust gases. Generally, for the fuel gas analysis, a chemist's services are needed. Strictly speaking, a fair sample of the gas used during the test should be analyzed, but when the fuel is such as city illuminating gas which does not vary much from day to day, an average analysis is fairly representative and may be used without considerable error.

The table, page 165, gives the average composition of the principal gas engine fuels.

The exhaust gas analysis may be made with the Orsat apparatus. The chief constituents are CO_2, O_2, and N_2, but an analysis for CO should not be omitted. A determination of H_2 is desirable, but not necessary if the CO is less than 1 per

* See also Appendix B, items 25 and 47

cent, as it is in proper operation. Since the CO is the least readily burned of the fuel gas constituents, it is a fair assumption that the hydrogen and hydrocarbons are completely burned if the CO is nearly all burned, and experiment bears out this assumption.

Knowing the fuel and exhaust gas analyses, the weights of gases in the exhaust pipe per cubic foot of fuel supplied may be calculated from the formulas deduced under Test 40 (*b*). This division should be re-read and thoroughly understood.

Sampling. For a chemist's analysis of the fuel gas, it is best to take a continuous sample covering the whole time of the test. This can be done by connecting a 5-gal. flask with rubber tubing to the gas main and syphoning water from the flask at a slow and uniform rate thus drawing in the sample. Sampling for heat value determination may be made as described on page 178, Test 39.

For throttling engines, the exhaust gas may be sampled as explained on page 193, Test 40. Hit-and-miss engines, it should be remembered, occasionally discharge air into the exhaust pipe which has not mixed with fuel. This air carries a certain amount of heat, though less than that carried by burned gases, and for this reason should be counted separately. It is possible to make an arrangement in the exhaust pipe by which the discharge from a missing stroke is by-passed and its temperature and quantity measured separately from the main exhaust. As this is a difficult matter, results may be obtained by averaging in the air from a missing stroke with the working exhaust, the sample then being taken as for a throttling engine. A thermometer in the exhaust pipe will show a mechanical average of the variable temperature which may be used for calculating the heat lost. When the exhaust from a hit-and-miss engine is sampled in this way, the excess coefficient cannot be figured from the analysis without first applying a correction for the air taken in by the engine, but not used to mix with the fuel.

Since both the fuel and exhaust gases are generally under pressure, it is easier and more accurate to obtain samples than in boiler work.

Duration of Test. This should be several hours, preferably, but, if on a small engine, a test of only one hour long will give valid results if the engine is previously run for fifteen or twenty minutes under the conditions to be maintained during the test. The readings to be mentioned later should be taken sufficiently often to obtain a fair average. The useful horse-power should be maintained at a constant value and all other conditions kept as uniform as possible. The test results are not valid unless the gas consumption is uniform as shown by meter readings taken at equal time intervals.

To make the methods clear, the measurements from a gas engine test will be worked through for the various results. Table No. 1 gives these measurements, and also the notation used in the formulas. The fuel used was illuminating gas manufactured in Syracuse, an average analysis of which is given in columns (1) and (2) of Table No. 2, page 294.

(a) Fuel Consumption, Cubic Feet of Gas per Brake horse-power-hour. Since the weight of fuel in a cubic foot varies with the pressure and temperature, the gas meter readings should be referred to standard conditions, namely, a pressure of 29.92 ins. of mercury and a temperature of 32° F. The following formula may be used. (See Table No. 1 for notation.)

$$V' = 16.4 \frac{B + \frac{P}{13.6}}{T_f + 460} \times V.$$

P and T_f necessitate the use of a water manometer and a thermometer at the point in the gas main where the gas meter is placed. For small size engines, the meter may be the usual gas meter, or a gasometer, but for large sizes, meters of the pitot, venturi, or orifice type may be used. When the engine is so large as to

TABLE 1

GIVING DATA FROM A GAS ENGINE TEST AND NOTATION USED IN FORMULAS

B.h.p. = brake horse-power	=25
I.h.p. = net indicated horse-power	=30.1
F.h.p. = friction horse-power	
V = number of cubic feet of fuel gas per hour, by meter	=414
V' = ditto, corrected to absolute pressure of 29.92 ins. of mercury and 32° F.	
B = Barometer reading, inches of mercury	29.5
P = pressure of fuel gas above atmosphere, at meter, inches of water	4
F = fuel consumption, standard cubic feet of gas per brake horse-power-hour	
T_f = temperature of fuel gas at meter, degrees F.	=68
t_j = temperature of ingoing jacket water, degrees F.	=61
T_j = temperature of outgoing jacket water, degrees F.	=133
T_e = temperature of exhaust gases, degrees F.	=750
t = temperature of air near engine, degrees F.	=70
W_j = weight of jacket water, pounds per hour	=1210
V_d = volume of dry exhaust gas from the combustion of 1 cu. ft. of fuel, cubic feet per cubic foot	
W_d = weight of dry exhaust gas from the combustion of 1 cu. ft. of fuel, pounds per standard cubic foot	
W_v = weight of water vapor, ditto	
R = ratio by volume of air consumed to fuel gas	
X = excess coefficient	
c, h, g = from sums of columns (3), (4), (5), Table 3, respectively.	

VOLUME ANALYSIS OF EXHAUST GAS, PER CENTS

D = carbon dioxide, CO_2	=9.0
O = oxygen, O_2	=8.8
M = carbon monoxide, CO	=0.5
H = hydrogen, H_2	Not analyzed.
N = nitrogen, N_2	=81.7

be inconvenient to brake, the useful horse-power may be approximated by subtracting from the indicated horse-power at the running load its value when the engine is running entirely free.

Applying the data of Table No. 1, we have,

$$V' = 16.4 \times \frac{29.5 + \frac{4}{13.6}}{68 + 460} \times 414 = 385 \text{ cu. ft. per hour.}$$

TABLE 2

ANALYSIS OF FUEL GAS USED DURING TEST, AND COMPUTATION OF QUANTITIES USED IN FIGURING RESULTS

(1) Constituents of Fuel.	(2) Per Cent by Volume.	(3) Subscript of $C \times (2)$.	(4) Ditto of H.	(5) Ditto of O.	(6) Higher Heat Value of (1)	(7) $\frac{(6) \times (2)}{100}$
CO	23.5	23.5	....	23.5	342	80.5
H_2	36.4	....	72.8	....	348	127
CH_4	15.3	15.3	61.2	....	1065	163
C_2H_4	6.9	13.8	27.6	....	1680	116
C_6H_6	3.9	23.4	23.4	....	4000	156
O_2	1.4	....	....	2.8	....	.....
CO_2	4.2	4.2	....	8.4	....	....
N_2	8.4	....	....	....	....	....
No. of Atoms		$80.2 = c$	$185.0 = 2h$	$34.7 = 2g$		643 =
No. of Molecules or Mol-volumes			$h = 92.5$	$g = 17.35$	higher heat value of fuel	

From which the fuel consumption is

$$F = \frac{385}{25} = 15.4 \text{ standard cubic feet per brake horse-power-hour.}$$

(b) Heat Supplied. Since the heat balance is based upon one cubic foot of the fuel gas, the heat supplied is its heating value. It is preferable to use a calorimeter for this determination, such as the Junker, but if one is not available a working value may be obtained by calculation from the fuel gas analysis, Test 39 (*c*). The calculation is made in column 7 of Table 2.

The gas engine code of the A.S.M.E. specifies that the "higher" heat value be used, that is, the heat obtainable from the fuel including the latent heat of vaporization of the water formed by combustion of the hydrogen. Some authorities contend that the lower heat value is more logical to use in this connection since the gas engine cannot avail itself of this

latent heat, but for the sake of uniformity the recommendations of the code will be followed.

(c) **Heat Converted into Useful Work.** The heat equivalent of one brake horse-power-hour is 2545 B.t.u. Since the fuel consumed to produce this power is F cu. ft., the heat equivalent of the useful power per cubic foot of fuel is

$$\frac{2545}{F}.$$

Applying the data of the example this is,

$$\frac{2545}{15.4}=165 \text{ B.t.u.}$$

Dividing this by the heat value gives 25.6 per cent. This is the thermal efficiency.

(d) **Heat Lost to Mechanical Friction.** In power units,

$$\text{F.h.p.}=\text{I.h.p.}-\text{B.h.p.}$$

To base this upon the heat of one cubic foot of fuel, a formula derived the same as the one last given is used, namely

$$\text{Friction loss, B.t.u.}=\frac{2545 \text{ F.h.p.}}{V'}.$$

Applying the data of the example, this is

$$\frac{2545(30.1-25)}{385}=33.7 \text{ B.t.u.}$$

or

$$33.7\div 643=5.2 \text{ per cent.}$$

(e) **Heat Lost to Jacket Water.** This is the heat absorbed in a given time divided by the fuel used in the same time. Considering hourly quantities,

$$\text{Jacket loss}=\frac{W_j(T_j-t_j)}{V'}.$$

The weight W_j may be measured by a barrel or tank on a platform scales. The temperatures should be read with two thermometers; one in the discharge tank (provided the piping is arranged to discharge without material radiation loss), the other in a thermometer well in the inlet pipe or in a receptacle drawing water from a tap in the main situated similarly to the jacket as to temperature.

Applying the data of the example,

$$\text{Jacket loss} = \frac{1210(133-61)}{385} = 226 \text{ B.t.u.}$$

or

$$226 \div 643 = 35.2 \text{ per cent.}$$

(f) Volume of Dry Exhaust Gas per Cubic Foot of Fuel Gas. This quantity is to be used in the various calculations from the exhaust gas analysis. Under Test 40 (*b*) there is deduced the formula,

$$V_d = \frac{c}{D+M},$$

c is found as in column 3 of Table 2. The subscript of the C in each fuel gas constituent shown in column 1 is multiplied by the volume percentage of that constituent as shown by column 2. Column 3 gives these products, the sum of which is the value of c.

For the data of the example

$$V_d = \frac{80.2}{9.0+0.5} = 8.44 \text{ cu. ft.}$$

(g) Loss of Heat to Dry Exhaust Gases. This determination involves the measurement of the temperature rise of the fuel and air mixing with it, and of the weight of the dry exhaust gases. For the specific heat, an average value, 0.245, may be used with sufficient accuracy. Then

Loss to dry exhaust gases $= 0.245\, W_d(T_e - t)$.

Under Test 40 (b) there is deduced the formula,

$$W_d = \frac{V_d}{9000}\left\{ 11D + 8O + 7(M+N) \right\},$$

by which the weight of the dry exhaust gas may be found from its analysis.

The temperature in the exhaust pipe should be taken by a high reading thermometer (1000° F.) inserted in a well filled with fine, dry sand, located as near the engine as possible.

Applying the data of the example,

$$W_d = \frac{8.44\{11\times 9 + 8\times 8.8 + 7\times(0.5+81.7)\}}{9000},$$

$$= 0.703 \text{ lbs. per cu. ft. of fuel.}$$

and loss to dry exhaust gas $= 0.245 \times 0.703 \times (750-70)$

$$= 118 \text{ B.t.u.}$$

or $$118 \div 643 = 18.3 \text{ per cent.}$$

(h) Loss of Heat to Water Vapor in the Exhaust. Since the higher heat value of the fuel is used, the latent heat of the water vapor is to be included. The small amount of water vapor brought in as humidity of the air and fuel may be ignored, particularly as its latent heat is not added by the fuel. This leaves only the water from the combustion of hydrogen. A formula for the weight of vapor so formed from one cubic foot of fuel gas is deduced under Test 40 (b). Combining this with the expression for total heat of steam under the given conditions (see page 143), we have,

Loss of heat to water vapor $= W_v \times$ (total heat of the vapor above room temperature)

$$= 0.0005h\ (1090 + 0.46T_e - t).$$

The value of h is found similarly to c, as shown by column 4, Table 2.

Applying the data of the example,

$$\text{Loss to water vapor} = 0.0005 \times 92.5 \times (1090 + 0.46 \times 750 - 70)$$
$$= 64.3 \text{ B.t.u.}$$

or

$$64.3 \div 643 = 10 \text{ per cent.}$$

(i) Heat Lost by Unburned Fuel Gas. The volume of unburned CO per cubic foot of fuel gas, according to Test 40 (*b*), is $V_t = V_d M \div 100$. As the heat value of CO is 342 B.t.u. per cubic foot, the

$$\text{Loss to unburned CO} = 3.42\ V_d M.$$

Similarly, if H per cent of hydrogen is found in the exhaust,

$$\text{Loss to unburned } H_2 = 3.46\ V_d H,$$

the higher heat value of hydrogen being 346 B.t.u.

These two expressions may be combined by assuming an average heat value, 344, to apply to both CO and H_2. Then

$$\text{Loss to unburned CO and } H_2 = 3.44\ V_d(M + H).$$

If hydrogen has not been analyzed for, H may be omitted.

Applying the data of the example,

$$\text{Loss to unburned fuel} = 3.44 \times 8.44 \times 0.5 = 14.6 \text{ B.t.u.}$$

or

$$14.6 \div 643 = 2.3 \text{ per cent.}$$

(j) Heat Lost to Radiation, etc., is found by subtracting from the heat value of the fuel the other quantities of the heat balance (*c*) to (*j*), or by subtracting from 100 per cent these quantities expressed in per cents. The unaccounted for heat includes that used for pumping the fuel mixture into the cylinder if the area of the lower loop has been subtracted from the upper of the indicator diagrams. This quantity could be separately measured,

but it is generally sufficient to include it in the friction loss as is done when the lower loop is altogether ignored, or to list it under radiation.

Applying the data of the example,

$$\text{Loss to radiation, etc.} = 643 - 165 - 33.7 - 226 - 118 - 64.3 - 14.6 = 21.4 \text{ B.t.u.},$$

or $$21.4 \div 643 = 3.4 \text{ per cent.}$$

(k) Cubic Feet of Air per Cubic Foot of Fuel Gas. This may be figured from the formula deduced under Test 40 (*b*),

$$R = \{V_a(D + O + 0.5M) + 0.5h - g\} \div 21$$

in which g is found under column 5, Table 2.

Applying the data of the example,

$$R = \{8.44 \times (9.0 + 8.8 + 0.5 \times 0.5) + 0.5 \times 92.5 - 17.35\} \div 21 = 8.63 \text{ cu. ft.}$$

(l) The Excess Coefficient. The following formula may be used.

$$X = \frac{21R}{c + 0.5h - g}.$$

Applying the data of the example,

$$X = \frac{21 \times 8.63}{80.2 + 0.5 \times 92.5 - 17.35} = 1.66.$$

Problem 60₁. The heat balance is found for a number of runs at different loads, and the items plotted thus: Eff. vs. B.h.p., Eff.+Friction vs. B.h.p., Eff.+Friction+Jacket loss vs. B.h.p., etc. Sketch the resulting curves in what you would expect to be their form and give reasoning.

Problem 60₂. If the air used in the test example given above was saturated with humidity, what would be the per cent loss to the heat carried away by this vapor? *Ans.*, 0.5%.

61. * Economy Test of an Automobile Engine

Principles. Engines of this type are variable speed. A complete test will therefore consist of enough runs at different speeds to cover the working range. As the rotative speeds are high, these engines cannot be indicated except with the optical indicator. The exhaust gas analysis cannot always be relied upon for the determination of the losses, because irregular firing, incomplete combustion, or lubricating oil in the exhaust will make erroneous the percentages as usually found. If conditions are such that the exhaust gas analysis is valid, the methods of the preceding test and the formulas for coal combustion, Test 40 (*b*), are applicable. In these formulas, $C_g = C_t =$ weight, in pounds, of the carbon in the fuel per pound of fuel; since all the carbon is gasified. In the formula for the weight of water vapor in the exhaust, the value of m, moisture in the fuel, drops out. The results from the formulas will be based on 1 lb. of fuel.

American gasoline, specific gravity about 0.7, contains roughly 83 per cent of carbon, 15 per cent of hydrogen, and less than 1 per cent of oxygen, by weight. Kerosene is a little higher in carbon and lower in hydrogen. The higher heat value of gasoline is about 20,000 B.t.u.; the lower, about 19,000 B.t.u. per pound. The heat values of kerosene per pound are practically the same, but based on the gallon they are higher, the specific gravity of kerosene being 0.8 or more.

(a) **Determination of Torque and Brake Horse-power at Various Speeds.** The ordinary Prony brake is difficult to use when it is sought to maintain a uniform load long enough for a fuel consumption test, because of the lack of uniformity of the frictional resistance. Fan brakes have been much used for the purpose, but they are not strictly reliable. Probably the best results ensue from the use of a well designed water brake or an electric dynamometer of the Sprague type.

* See Appendix B, item 47.

There are several combinations of variables possible, making it requisite to decide upon a definite independent variable. For example, a series of tests may be run at different speeds, and always a wide open throttle (that is, the valve controlling the amount of air fuel mixture). This will result in maximum torque at each speed and increasing horse-power up to a certain limiting speed. Or, a similar series of tests may be run at a partial throttle opening. Or, the torque and speed may be kept constant (corresponding to average road conditions) and the air-fuel ratio varied, or changes in the spark timing made, or temperature of inlet or outlet water, inlet air, etc. Of course, when any such set of tests is decided upon it is very important that all other quantities be kept as constant as possible, such as temperatures, mixtures, carburetor adjustments, and so on. In all cases curves of the results obtained should be plotted.

The brake horse-power of automobile engines is often estimated by empirical formulas, especially for the purpose of rating. The Association of Licensed Automobile Manufacturers uses the formula

$$\text{B.h.p.} = D^2 \times \text{number of cylinders} \div 2.5,$$

D being the bore in inches. This is referred to as the A.L.A.M. rating and applies to four-cycle engines at an assumed piston speed of 1000 feet per minute.

(b) **Brake Mean Effective Pressure.** Because of the difficulty of determining the real mean effective pressure of a high speed engine, the following expedient is used. Since the B.h.p. $=$ Mech. Eff. $\times P\ L\ A\ N \div 33000$; the product, **Mechanical Efficiency $\times$ M.e.p.** $=$ B.h.p. $\times 33000 \div L\ A\ N$. This product is measurable, as shown by the terms it is equal to, and is called the "Brake Mean Effective Pressure."

(c) **Fuel Consumption, Gallons or Pounds per Brake Horse-power-hour.** The general principles of this determination are the same as for other fuel consumption tests except as regards the measurement of fuel. For this purpose a calibrated tank may be used, containing the oil with which to supply the engine. In some cases, it is desirable to keep the head of oil on the float valve of the carburetor practically constant. An arrangement may then be used similar, on a small scale, to that for measuring feed water to a boiler, Fig. 100. Another plan is to place the supply vessel on a scales and to syphon the fuel from it.

(d) **Thermal Efficiency and Heat Balance.** These subjects are dealt in the same manner as in Test 60, formulas to be used being given under Test 40 (b).

62.* Economy Test of a Gas Producer

Principles. The distribution of the heat of the coal consumed by a producer is in four directions. First, the useful heat, appearing as the calorific value of the dry, cool, gas; second, the heat carried by the gas as sensible heat and as latent heat of the excess steam; third, the heat lost to good fuel removed with ash; fourth, the heat lost through deposits of tar and soot, absorption and leakage of gases, and radiation from the producer.

The selection of methods of measurement of these heat quantities is governed somewhat by the type of producer and fuel, whether pressure or suction, anthracite or bituminous. In any case, it is necessary to measure the coal and ash during a test of sufficient duration, and to provide a proximate analysis of the coal and a chemist's analysis of an average sample of the gas. The gas usually contains between 10 and 20 per cent of hydrogen, 1 to 3 per cent of marsh gas, and a small amount of olefiant gas, in addition to CO_2, CO, and N_2. These make it not easy

* See also Appendix B, items 24 and 46.

to analyze, although it may be done without much labor if the apparatus is available.

The calorific values of both coal and gas may be calculated from their analyses.

The volume of gas generated may be figured from the gas analysis with acceptable accuracy in most cases, thus dispensing with the bothersome meter question. This calculation depends upon the assumption that all the carbon gasified from the fuel appears in the gas analysis. It is therefore approximate since the condensible hydrocarbons distilled from the coal throw down some carbon in the form of tar, and some disappears as soot. In the case of anthracite gas, or well fixed bituminous, this is negligibly small since, generally, it does not involve more error than is inherent to the coal measurement.

The largest error likely to occur is in the measurement of the coal consumed, in that the producer may contain a very different amount at the end of the test from its contents at the beginning. To avoid this, the conditions of the producer at **starting and stopping** should be noted and made in all respects as nearly the same as is possible. Starting and stopping should be be timed just after a regular cleaning. **The duration of the test** should be long enough to reduce the probable error caused by the variation in fuel content of the producer to a small percentage of the total coal fired, which, in general, will not be less than twelve hours. If the weight of each charge of coal is plotted as a total against the time at which it is consumed, the resulting curve will show by its straightness the uniformity of coal consumption. For such uniformity all external conditions must be kept as constant as possible and the producer should be run for some hours under the required conditions before the main test is started.

The ash and refuse should be collected at regular cleaning intervals and weighed after drying. It is best to secure a sample of the total refuse just before weighing and to determine its

moisture and carbon as in the proximate analysis of the coal. It is often awkward to dry all the refuse thoroughly, but, if the percentage of moisture is known, the weight of dry refuse may be readily figured.

Sampling of the coal and ash should be done as for a boiler test. Sampling of the gas should be the same as for the fuel gas during a gas engine test (see page 178). The gas should be collected before entering the scrubber, and after leaving it if it is desired to find the effect of absorption or leakage.

The weight of steam furnished to a suction producer exclusive of that due to vapor in the air and moisture in the coal may be found by supplying the vaporizer with water from a calibrated vessel and collecting and weighing the overflow. For a pressure producer, an orifice or nozzle method may be used to meter the steam, or it may be measured by weighing the feed water evaporated by the auxiliary boiler.

The calculations from the test of an anthracite suction producer plant will first be given and then the modifications necessary for other types. Tables 1 and 2, pages 305 and 306, give the required test data together with the notation used in the formulas. The data are part of the full observations of a test reported in the Journal of the A.S.M.E., December, 1909, for which the ultimate as well as the proximate coal analysis and calorimetric determinations of fuel and gas were made, and the gas metered. The reader may thus compare the results by the approximate methods to be presented with the more exact results of the report.*

All the items of the heat balance will be figured on the basis of 1 lb. of dry coal. The calorific value of the dry coal will thus

* Note that these results are given for the standard gas condition of 62° F. and 30 ins. of mercury, whereas the standard here used is for 32° F. and 29.92 ins. The volume per pound of the latter is 0.943 times that of the former. Also the specific heats of the gases are different, the values here quoted being by later authorities.

TABLE 1

GIVING DATA FROM A SUCTION GAS PRODUCER TEST AND NOTATION USED IN FORMULAS

Total weight of coal charged, pounds	=798
Total weight of ash and refuse, pounds	= 85
Total weight of water used in vaporizer	=268

PROXIMATE ANALYSIS OF THE COAL. PER CENTS BY WEIGHT OF THE COAL AS FIRED

Moisture	= 2.75
Volatile matter	= 6.00
Fixed carbon	=78.45
Ash	=12.8

m, vm, fc = weights of moisture, volatile matter, and fixed carbon per pound of dry coal, respectively.

Weight of carbon in 1 lb. of dry refuse, pounds = 0.388

T_p = temperature of gas leaving producer, degrees F. = 1108

t = temperature of fire room, degrees F. = 82

C_t = total weight of carbon in 1 lb. of dry coal, pounds.

C_a = weight of carbon wasted in ash, per pound dry coal, pounds.

C_g = weight of carbon gasified per pound dry coal, pounds.

c, p = sums of columns 3 and 5, Table 2, respectively.

V = volume of dry gas generated per pound of dry coal, standard cubic feet.

W_p = weight of the producer gas per pound of dry coal, pounds.

W_a = weight of the air furnished per pound of dry coal, pounds.

W_s = weight of steam furnished per pound of dry coal, from the vaporizer, pounds.

w_s = weight of steam not dissociated per pound of dry coal, pounds.

be the heat supplied; the sensible heat carried in by air, water, etc., being neglected.

The principles upon which most of the calculations depend are given under Test 40 (*b*), coal combustion, and should be thoroughly understood.

(a) Quantities to be Found Prior to Calculation of the Results. The proximate analysis of the coal should be based upon its dry weight. This calculation is the same as for Test 54 (*c*), as is that for C_t, the total carbon per pound of dry coal; since the same

Table 2

ANALYSIS OF GAS GENERATED DURING TEST, AND COMPUTATION OF QUANTITIES USED IN FIGURING RESULTS

(1) Constituents of Gas.	(2) Percentage by Volume.	(3) Subscript of C × (2)	(4) Molecular Wt. of (1).	(5) (2) × (4)	(6) Higher Heat Value of (1).	(7) $\frac{(6) \times (2)}{100}$
CO	27.0	27.0	28	756	342	92.4
H_2	10.4	.....	2	20.8	348	36.2
CH_4	1.8	1.8	16	28.8	1065	19.2
C_2H_4	0.0	0	28	0	1680	0
O_2	0.2	.....	32	6.4	....	
CO_2	4.2	4.2	44	184.8	....	
N_2	56.4	....	28	1579.2		
		$c=33.0$		$p=2576.0$		147.8 = higher heat value of gas

fuel analysis has been assumed. The methods of finding C_a and C_g are also the same as given under Test 54 (*c*). For the data given by Table 1,

$$\text{Total weight of dry coal consumed} = 798 - 0.0275 \times 798$$
$$= 776 \text{ lbs.}$$

and
$$C_a = \frac{85 \times 0.388}{776} = 0.042 \text{ lb.}$$

The carbon gasified is

$$C_g = C_t - C_a = 0.816 - 0.042 = 0.774 \text{ lb. per pound of dry coal.}$$

The volume of dry gas generated per pound of dry coal, under standard conditions of pressure and temperature, may be found from a formula derived similarly to that for V_a, page 198. For producer work $D+M$ becomes c since carbon may

exist in other gases than CO_2 and CO. c is found as shown in column 3 of Table 2. Then

$$V=\frac{2980\ C_g}{c}.$$

For the given data,

$$V=\frac{2980\times 0.774}{33}=69.9 \text{ cu. ft. per lb. dry coal.}$$

The weight of dry gas per pound of dry coal may be obtained from its volume if the weight of a cubic foot of the gas is known. For the latter quantity, the method of finding the density of a gas described on page 197 may be used. The weight of a mol of the producer gas is $p\div 100$, this being its average molecular weight as found in column 5, Table 2. Since the volume of one mol of a gas is 359 cu. ft., its weight in pounds per cubic foot, or

$$\text{Density of dry producer gas}=\frac{p\div 100}{359}.$$

For the given data,

$$\begin{aligned}\text{Density of producer gas} &= 25.76\div 359\\ &= 0.0721 \text{ lb. per cubic foot,}\end{aligned}$$

from which the weight of the dry producer gas per pound of dry coal is

$$W_g=0.0721\times 69.9=5.04 \text{ lbs.}$$

The volume of gas may also be obtained by metering it through the whole test, but the method outlined above will generally be found sufficiently accurate if reasonable care is taken in the sampling and analysis of the gas.

The weight of air supplied to the producer per pound of dry coal may be found from an expression similar to that for W_g for

coal combustion, page 199; the quantity $D+M$ being replaced by c. The formula then is,

$$W_a=\frac{3.04\times N_2\times C_g}{c},$$

N_2 being the per cent by volume of nitrogen in the gas as given in column 2, Table 2. For the given data,

$$W_a=\frac{3.04\times 56.4\times 0.774}{33}=4.02 \text{ lb.}$$

The weight of steam supplied by the vaporizer per pound of dry coal is found by dividing the total weight of water to the vaporizer by total weight of dry coal. For the given data,

$$W_s=\frac{268}{776}=0.345 \text{ lb.}$$

Coming now to the heat balance, the first item is

(b) Useful Heat in the Cool Gas. This is exclusive of sensible heat if the gas is to be used for power, and is due to its calorific value when completely burned. The higher heat value will be used. Since all items of the heat balance are based on 1 lb. of coal, the useful heat may be found by multiplying the volume of gas per pound of coal, V, by the higher heat value in B.t.u. per standard cubic foot. The heat value may be figured as in column 7 of Table 2. This gives the heat values of the gas constituents, according to their amounts in one cubic foot, the sum of which is the required heat value. For the given data,

$$\text{Useful heat}=69.9\times 148=10{,}300 \text{ B.t.u.}$$

The heat value of the dry coal was found to be 13,040 B.t.u., so the useful heat is 10,300÷13,040=79.0 per cent. This is the " cold gas " efficiency of the producer.

(c) Sensible Heat in the Dry Gas Leaving the Producer is equal to

$$W_p \times \text{specific heat} \times (T_p - t).$$

The weight of the dry gas, W_p, has been previously figured.

The specific heat of producer gas, varies greatly with the hydrogen content. Except for hydrogen the average specific heat of all the gases, exclusive of steam, may be taken as 0.26 for a temperature of 1100° or 1200° F. The specific heat of hydrogen for these temperatures is about 3.72, so that even though its per cent by weight is always small the effect of hydrogen upon the average specific heat of producer gas is very marked and varies with the amount of hydrogen present. The curve, Fig. 104, gives the average specific heat of normal producer gas with various percentages of hydrogen. For the given data ($H_2 = 10.4$ per cent) it is seen to be 0.288.

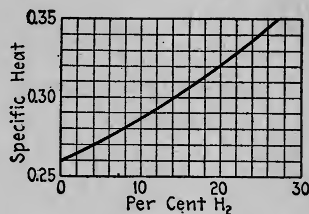


FIG. 104.—Specific Heat of Producer Gas.

The temperature of the gas should be taken near the outlet at the producer with a suitable pyrometer.

For the given data,

$$\text{Sensible heat} = 5.04 \times 0.288 \times (1108 - 82) = 1490 \text{ B.t.u.}$$

or $\quad 1490 \div 13{,}040 = 11.4$ per cent.

(d) Heat Lost by Excess of Steam to the Producer. This is estimated from the weight of steam found in the gas per pound of dry coal, and equals

$$w_s \times \text{total heat per pound of the steam above feed temperature.}$$

The total heat of the steam may be taken with sufficient accuracy for this calculation as

$$1000+0.5\times T_p.$$

For a more exact value, the expression quoted on page 143 may be used.

To find w_s several methods may be used. It may be measured directly by drawing a sample of the producer gas through a tube of calcium chloride, the increase of weight of which gives the moisture absorbed. The gas is syphoned into a container of known size so that its volume, and hence its weight, may be obtained. w_s may also be figured from the hydrogen content of the coal and the gas. An easier method is based upon the fact that the sum of the weights of the gasified carbon, air, and water vapor fed into the producer must equal the weight of the producer gas including moisture. Hence,

$$C_g+W_a+W_s+m+\text{vapor brought in with air}-W_p=w_s.$$

The air vapor may be neglected here, although it may be a large percentage of the total steam, since its latent heat is not added by the coal. Then, for the given data,

$$w_s=0.774+4.02+0.345+0.0283-5.04=0.137 \text{ lb.},$$

from which the heat lost to steam is

$$0.137\times(1000+0.5\times1108)=213 \text{ B.t.u.}$$

or

$$213\div13{,}040=1.6 \text{ per cent.}$$

(e) Heat Transferred to Scrubber Water. Items (*c*) and (*d*) together equal the heat given to the scrubber plus the heat radiated between the scrubber and producer outlets. For purposes of comparison, or to get an estimate of the amount of

sensible and latent heat in the gas without measuring it as just described, the heat added to the scrubber water may be determined. The water may be measured by a meter, and its temperature as for the jacket water of a gas engine. In the test quoted the total weight of scrubber water was 22,200 lbs. and its temperature rise was 45.8° F. Hence, the heat given to the scrubber per pound of dry coal is,

$$\frac{22,200}{776} \times 45.8 = 1310 \text{ B.t.u.}$$

(f) The Heat Lost to Carbon in the Ash. Taking the heat value of carbon as 14,600 B.t.u., this loss, per pound of dry coal is

$$14,600 \times C_a.$$

For the given data,

$$14,600 \times 0.042 = 613 \text{ B.t.u.}$$

or

$$613 \div 13,040 = 4.7 \text{ per cent.}$$

(g) Radiation from the Producer and other Losses. These may be grouped in one item by subtracting the previously calculated heat balance quantities from the heat supplied, or some of them may be separately determined. The loss by leakage of air in the case of a suction producer plant may be found by comparing the useful heat, as herein calculated, before and after the dilution. In a pressure producer plant, any leakage is apt to be detected; the only way to measure it is by metering the gas since leakage does not alter its composition. The loss due to tar and soot may be found by a special analysis of the gas as it leaves the producer.

Grouping these losses for the given data, we have

$$\text{Radiation, etc.} = 100 - (79.0 + 11.4 + 1.6 + 4.7) = 3.3 \text{ per cent.}$$

(h) **Other Efficiencies and Results.** **The "hot gas" efficiency** is often expressed when the gas is used for heating. In this case the sensible heat is also "useful." If, then, the first three items of the heat balance in per cent are added, the result will be the hot gas efficiency. For the given data, this is

$$79.0+11.4+1.6=92.0 \text{ per cent.}$$

The "efficiency based on combustible," corresponding to the boiler efficiency so known, is the efficiency that would be obtained if there were no loss of fuel in the ash, and equals

$$\text{Eff. based on coal} \div \left(1-\frac{C_a}{vm+fc}\right),$$

the quantity in the parenthesis being the efficiency of the grate, or cleaning. For the given data, this efficiency equals

$$79.0 \div \left(1-\frac{.042}{0.869}\right)=82.9 \text{ per cent}$$

for the cold gas.

For purposes of comparison of tests with different kinds of fuel, it is useful to obtain the **volume of gas per pound of combustible.** This equals $V \div (vm+fc)$.

In **capacity** tests, the output may be measured in cubic feet per hour by multiplying V, as herein calculated, by the weight of dry coal used per hour.

For **pressure producers** it is necessary to account for the coal used in the auxiliary boiler to make steam. This may be included in the total coal charged against the producer, and another item added to the heat balance to account for the heat lost in the steam making process. Or we may subtract from the volume of gas generated an amount equivalent to the steam supplied, which does not seem to be so logical a process.

For **bituminous producers,** if much carbon disappears as tar, it is necessary either to meter the gas to get V, or to determine the amount of carbon going to tar from 1 lb. of dry coal, by analysis of a sample extracted from a measured volume of gas. From this result a more correct value of the carbon gasified is obtainable.

The heat balance may be extended to cover the distribution of heat in a gas engine supplied by the producer, by the methods indicated under Test 57 covering this subject. In this case the gas need not be metered, since the fuel quantities are measured as coal, and the heat quantities should be based upon 1 lb. of dry coal instead of 1 cu. ft. of gas. The exhaust losses as calculated under Test 57 may be readily changed to the coal basis by multiplying them by V as herein obtained.

Problem 62_1. Using the data of the test example, what is the percentage of heat radiated from the scrubber and immediate piping?

Problem 62_2. Assuming that the engine of the example under Test 60 was operated with gas from the producer of the example of Test 62 with the same engine heat balance items, calculate these items in per cents of the heat value of the dry coal.

63. Test of a Refrigeration Plant *

(Ammonia Compression System)

Principles. The student, it is assumed, is informed upon the mechanical and thermal features of the ammonia refrigerating machine, descriptions of which are available in numerous treatises.

Following are given definitions of the quantities usually sought in tests of refrigerating machinery.

Refrigerating Effect is the amount of heat abstracted by the ammonia from the cooling medium, as brine, expressed in B.t.u. per unit of time (minute, hour or twenty-four-hour day). This includes the waste cooling effect due to heat transferred from

* This and the two following tests are enlarged from a series by the author, in *Power*, Sept. 19, Oct. 10, 1916, and April 10, 1917.

* See also Appendix B, items 34 and 58.

bodies it is not desired to cool—a waste necessarily ensuing from imperfect insulation, manipulation, etc.

The Unit of Refrigeration is a refrigerating effect of 288,000 B.t.u. per twenty-four hours, or 200 B.t.u. per minute. To obtain this effect by ice at 32° F., it would be necessary to melt 1 ton (2000 lbs.) of it in twenty-four hours, the latent heat of ice being 144 B.t.u. Units of refrigeration are therefore spoken of as tons more commonly than as B.t.u.

The Capacity of a plant equals the number of units of refrigeration it delivers: That is, the number of tons of 32° ice it would be necessary to melt per twenty-four-hour day to produce a refrigerating effect equivalent to that of the ammonia. Hence, this quantity is often referred to as "ice-melting capacity."

Ice-making Capacity is sometimes considered in the case of plants devoted exclusively to the manufacture of ice. Expressed in tons per twenty-four hours, this is equal to between one-half and eight-tenths of the ice-melting capacity previously defined, the diminution being due to the losses in the process of heat abstraction by the brine, manipulations of cans, and to the fact that the water must first be cooled to 32° and after freezing, be chilled below that point. Rating in terms of ice-making capacity is therefore only definite when the temperatures of water and finished ice are stated.

The Coefficient of Performance is a more correct term for what is sometimes miscalled the efficiency of a refrigerating system. This quantity is the ratio of the refrigerating effect in B.t.u. per unit of time to the heat equivalent to the indicated work done by the compressor in the same time; that is, it is the useful effect divided by the power put in, and in this respect is superficially an efficiency. The term efficiency generally has reference to the relation of an energy, after passing through a conversion, to its value at the source. In the case of refrigerating machinery the compressor energy is *not the source* of the useful effect. In reality, it is the condensing water that does the cooling, the com-

pressor being merely an auxiliary in the process. As the work done upon the ammonia is the paid-for item and as the refrigerating effect is the thing desired, it is useful to know the ratio of the two.

The Ideal Coefficient of Performance is the maximum that could be obtained under a given set of operating conditions if there were no losses. It is fixed by the operating temperatures of condenser and refrigerator and is equal to

$$P_i = \frac{460 + T_r}{T_c - T_r},$$

in which the numerator is the absolute temperature of the refrigerator, T_r is its temperature in degrees Fahrenheit, and T_c is the temperature of the condenser in the same units. It should be noticed that neither refrigerator nor condenser is ever at one uniform temperature in operation. To maintain the heat flow, the brine must always be a little warmer than the ammonia it boils, and the circulating water always a little cooler than the vapor it condenses. But, assuming ideal apparatus, the ammonia vapor could be worked in the refrigerator up to the temperature of the outgoing brine (just as in a perfect steam boiler the steam might be worked up to the temperature of the outgoing flue gases), and in the condenser the vapor could be worked down to the temperature of the outgoing water. If these temperatures be used in the calculation, then, the resulting coefficient of performance represents ideal conditions of the whole system, including heat transmission of refrigerator and condenser. If, however, the temperatures corresponding to the vapor pressures of the ammonia are used, the resulting coefficient is that of the system, exclusive of losses in heat transmission in refrigerator and condenser. Numerical examples of these two values of the ideal coefficient of performance will be cited later.

The Efficiency of the plant may be expressed as the ratio of the actual to the ideal coefficient of performance.

The Economy of compression plants is often stated in pounds of refrigerating effect (ice-melting) per pound of coal, on the assumption that a certain number of pounds of coal are consumed to make one indicated horse-power at the compressor.

Other Results should include the expense of condensing water (under certain conditions this may be as large as the cost of the fuel) and the cost of auxiliary power.

For complete tests, enabling a study of all the heat transfers, temperature and time-quantity determination should be made of both ammonia and cooling water.

The Duration of the trial should be at least twelve hours, and preferably twenty-four, to allow for the possible error at the finish of the test due to a different amount of heat being stored in the refrigerator, condenser, etc., from that contained at the beginning.

(a) Refrigerating Effect by Measurement of the Brine. Refrigerating effect in B.t.u. per minute, equals

$$R=W(T-t)C$$

in which

W = weight of brine in pounds per minute;
T = temperature of brine at inlet in degrees Fahrenheit;
t = temperature of brine at outlet in degrees Fahrenheit;
C = specific heat of brine.

For the determination of W the various methods of water measurement have been applied. Owing to the large quantity of brine circulated even in small plants, the method of direct weighing by tanks and scales is inconvenient. A modification of this consists in running the brine through two tanks, one above the other, the upper one having in its bottom a number of orifices through which the brine flows. The rate of flow varies with the height of the brine level in the upper tank. One of the orifices may be readily calibrated during the trial by collecting, weighing

and timing its discharge at various heads. The total brine flowing through all the orifices may then be found by multiplication.

When meters calibrated in gallons or cubic feet are used, a separate determination of the weight of the brine per gallon or cubic foot is necessary. This may be made with a hydrometer, care being taken to test a sample withdrawn near the meter and at approximately the same temperature as it has in the pipe.

Concerning temperatures T and t, as the range is only between 5 and 10°, finely graduated thermometers must be used. For not more than 2 per cent of error the graduations must be about 2 per cent of the temperature range—that is, between one-fifth and one-tenth degree. The thermometers should be inserted in wells filled with mercury or oil and should be located in the inlet and outlet pipes as near as possible to the cooler. The protruding portions of the wells should be well insulated. During the trial it is well to interchange the thermometers as a check on their accuracy.

The **Specific Heat of Brine** varies with its concentration, constituents and temperature. The last-named variation is comparatively small—about 0.05 per cent decrease of the specific heat for each degree decrease in temperature. As to the effect of the constituents, the presence in calcium chloride brine of manganese and sodium chloride in moderate quantities (say up to 20 per cent) does not affect the specific heat of the mixture materially. The effect of the concentration, however, is to lower the specific heat from unity (that of water) down to about 0.65 at a concentration corresponding to a specific gravity of 1.26.

The following formula for pure calcium chloride brine will give results for the specific heat of commercial calcium chloride brine to within 1 or 2 per cent of error, between the limits of −4 and 40° F., and specific gravities between 1.10 and 1.26:

$$C = 1.833 - 0.93G - 0.0005(32 - T_a).$$

In this formula* G is the specific gravity, and T_a is the average temperature of the brine in degrees Fahrenheit at inlet and outlet.

(b) **Refrigerating Effect by Ammonia Measurements.** The heat added to the ammonia in the brine cooler equals the heat lost by the brine. If the former quantity be found it is therefore a measure of the refrigerating effect, and we can put

$$R = A \times (H - h'),$$

in which R is as before, A is the number of pounds of anhydrous ammonia circulated per minute, and

H = the total heat of the ammonia per pound leaving the cooler,
h' = the heat of the liquid at the expansion valve.

To find A, see Test 64 (h).

H and h' are to be found from tables of the properties of ammonia,† or from the Mollier diagram for ammonia. To use these tables, the same data as for steam are necessary. For H the suction pressure must be read, together with the temperature, so that the heat of superheat may be calculated. The specific heat of superheated ammonia at low pressures may be taken as 0.51. For h', the head pressure may be referred to.

It is difficult to get the exact amount of superheat of the ammonia leaving the cooler, and also to maintain a uniform quantity of ammonia in the cooler to avoid the error of starting and stopping as in a boiler test. For these reasons the refrigerating effect as obtained by this method may be expected only roughly to check the result by the brine method which is therefore to be preferred.

When the cooling is done by the direct expansion of ammonia, there is no choice between these methods, as brine measurements are eliminated under such conditions. An approximation of the

* Deduced from the values given in Bureau of Standards Bulletin No. 135.
† See Appendix for properties of saturated ammonia.

refrigerating effect can be made by still another method which is also useful as a rough check of either (*a*) or (*b*).

(c) **Approximate Calculation of Refrigerating Effect.** Neglecting effects of radiation, etc., the heat added in the refrigerating plant equals the heat equivalent to the compressor work, plus the heat added to the ammonia, *R*. This sum equals the heat taken away, or the heat imparted to circulating water. By finding the heat taken up by the condenser water and jackets and subtracting from this the heat equivalent to the compressor work, we thus have a rough measure of the refrigerating effect.

The condenser water may be measured as under (*g*). Its temperature rise should be obtained by thermometers at inlet and outlet, from which data the heat removed may be calculated.

(d) **Ice-melting Capacity** is readily calculated from the value of the refrigerating effect *R* in B.t.u. per minute, by dividing by 200, the result being in tons per twenty-four hours.

(e) To obtain **the actual coefficient of performance**, it is necessary to indicate the compressor. The process here is similar to that for steam-engine trials and hardly needs comment other than that a special steel-lined indicator must be used with as short pipe connections as possible to avoid materially increasing the clearance space. Having obtained the indicated horse-power of the compressor, the coefficient sought is

$$P_a=\frac{R}{42.4\times \text{I.h.p.}}.$$

(f) **The Ideal Coffiecient of Performance** may be found from the average thermometer readings at the brine and condenser water outlets, as previously defined. Or, if vapor pressures are used, the corresponding temperatures may be found from the ammonia tables. When the plant is equipped with pressure gages bearing thermometric scales, the ammonia tables may be dispensed with.

As an illustration of the difference that may exist between the values of the ideal coefficient as obtained by the two methods previously discussed, consider the following data:

Suction pressure, 28-lb. gage; from tables, temperature, 15°; head pressure, 151-lb. gage; from tables, temperature, 84.5°; outgoing brine temperature, 28.9°; outgoing condenser-water temperature, 83.5°.

From these figures the ideal coefficient of the whole system is

$$\frac{460+28.9}{83.5-28.9}=8.95$$

and for the system, exclusive of condenser and refrigerator transmission losses, is

$$\frac{460+15}{84.5-15}=6.83.$$

(g) **Gallons of Cooling Water per Minute per Ton.** The measurement of the condenser water is simpler than that of the brine, the quantity being much less. Direct weighing or the usual forms of water meter may be employed, care being taken to select one appropriate to the conditions of flow, whether pulsating or steady. The number of gallons of cooling water per minute per ton of ice-melting capacity should be figured for purposes of comparison and also in order to add the cost of this item to the other costs to obtain the total. This item should include all cooling water used, such as compressor jacket water and water to steam condenser if one is used.

(h) **Overall Economy.** The performance of auxiliaries may be either estimated or measured. When they and the compressor are steam driven, it is well to measure the total steam supplied to the plant by the boiler-feed method or by a steam meter installed so as to avoid inaccuracies due to pulsating flow. The steam consumption of the compressor and of the whole plant may then be stated in pounds of steam per ton of ice-melting.

Ratings are generally based on head and suction pressures of 185 and 15.3 gage respectively, corresponding to temperatures of 96 and 0° F., respectively. Any departure from these conditions should be considered in making comparisons of test results.

Problem 63_1. What is the tonnage (ice-melting capacity) of a plant that circulates 6000 lbs. of brine (specific heat = 0.80) per hour at an average temperature fall of 15° F.? *Ans.* 6 tons.

Problem 63_2. What is the specific heat of a calcium chloride brine having a specific gravity of 1.2, if its temperature is 32° F.? If its temperature is 0° F.? *Ans.*, 0.717; 0.701.

Problem 63_3. Supposing 600 lbs. of anhydrous ammonia are circulated per hour at a head pressure of 160 lbs. gage, and a suction pressure of 20 lbs. gage, the superheat being 25°; what should be the refrigerating effect? *Ans.*, 29,600 B.t.u. per hr.

Problem 63_4. A refrigerating plant uses 14,220 lbs. of water per hour in the ammonia condenser, with average incoming and outgoing temperatures of 50° and 75°, respectively. The I.h.p. of the compressor is 20. Approximately what is the tonnage (ice-melting capacity)? *Ans.* 25.4 tons.

Problem 63_5. What is the actual coefficient of performance for the data of Problem 63_4? *Ans.* 6.

64. Test of a Refrigeration Plant.

(Ammonia Absorption System)

Principles. In some respects the results from and methods for the test of an absorption plant are identical to those applying to the compression system, for which see Test 63. The capacity in terms of ice-melting effect may be found in the same way, and the investigation of the condenser operation is the same. The actual coefficient of performance may not readily be obtained, since the energy imparted to the ammonia (barring the work of the ammonia pump, which is comparatively small) is in the form of heat instead of compressor work. But the ideal coefficient of performance may be determined as for a compression plant and may be useful for purposes of comparison of the two systems.

The principal economy result is the steam consumption expressed in pounds of steam per hour per ton of refrigerating

effect. This may be determined *in toto* for the plant or itemized as live steam to the pump and steam to the generator, the latter being subdivided into live and exhaust steam according to whether the one or the other or a combination is used. Under favorable circumstances the generator will use 30 lbs. of steam per hour per ton. The steam consumption of ammonia pumps is subject to wide variations due to their construction. Although the energy delivered is small compared with the heat transferred in the generator, a test of this pump should be included in economy trials because of the bearing of its performance on the steam consumption of the plant; and the pump test should preferably include a determination of the horse-power delivered.

In addition to these results it is desirable to ascertain the condition of the aqua ammonia and, if convenient, the amount of anhydrous circulated per unit of time. The former is expressed in terms of its concentration; that is, the part of a pound of ammonia in one pound of the aqua ammonia (more frequently expressed as a percentage). The concentration results show the working of the absorber and generator and enable calculations regarding the ammonia circulation.

(a) **Refrigerating Effect,** 2 methods. See Test 63, (*a*) and (*b*).

(b) **Ice-melting Capacity.** See Test 63, (*d*).

(c) **Ideal Coefficient of Performance.** See Test 63 (*f*).

(d) **Gallons of Cooling Water per Minute per Ton.** See Test 63 (*g*) in so far as condenser water is concerned. All other cooling water supplied, if in separate amounts, should be included in this total, by similar methods of measurement.

(e) **The Steam Consumed by the Generator** is readily ascertained by piping the generator-trap discharge into a weighing barrel. The barrel, to start with, should be about half full of cold water to prevent evaporation of the condensate from the trap. Quick emptying should be provided for. Weights should be recorded at uniform time intervals (to show uniformity of operation), from which data the rate in pounds per hour may be had.

Quality determinations of the steam are important, unless an efficient separator is installed to remove the moisture from the incoming steam, to the consideration of the results, as also are pressure readings.

If reduced live steam is used in addition to exhaust and it is sought to measure them separately, a steam meter may be applied to the live-steam line. The exhaust-steam quantity then equals the total trap condensate minus the meter reading. Measurement of the feed water, as for a boiler test, may also be resorted to, in cases where the plant may be isolated.

When the generator is supplied with exhaust steam from the pumps plus reduced live steam, the live steam may be found by subtracting from the generator trap discharge the steam consumed by the pumps. This last may be measured as follows:

(f) Steam Used by Ammonia and Brine pumps. A separate test should be made to determine this, but if conditions are not favorable, the steam required to drive the pumps may be estimated from the manufactuer's figures for their water rates and from the horse-power actually delivered. These remarks of course apply to steam pumps; where power pumps are installed, the energy may be considered in the list of costs.

(g) Work of the Ammonia-pump. It is preferable to find the net horse-power output rather than the indicated horse-power. From the data required for the net horse-power and from the number of pump displacements, the slip may be calculated as in ordinary pumping-engine trials.

The required formula for horse-power is

$$\text{Net horse-power} = \frac{144(P_1 - P_2)V}{33{,}000},$$

in which P_1 and P_2 are the gage pressures worked against in pounds per square inch (that is, of the generator and absorber, respectively) and V is the number of cubic feet of strong aqua circulated per minute.

The quantity of strong aqua V may be determined in several ways. A specially calibrated meter may be used on the discharge side of the pump; the aqua may be measured in calibrated tanks on the suction side; or it may be calculated from the amount of anhydrous circulating or from the quantity of weak aqua, the concentrations being known. For this last method (see (*j*)).

When a receiver is installed to supply the ammonia pump, it may be readily calibrated so that the volume contained, corresponding to any height of the liquid in the gage-glass, is known. Now, by cutting off the flow from the absorber into this receiver for a short interval, the rate, in cubic feet per minute, at which the strong aqua is pumped out, may be ascertained. If desired, two receivers may be arranged in parallel, thereby enabling continuous measurement.

(h) Weight of Anhydrous NH_3 Circulated per Minute. Volume measurements may be made in the same way as described under (*g*) for strong aqua. To convert the volumetric results into weights, a temperature or pressure reading also should be obtained to find the density of the liquid from the tables of properties of ammonia. When closely accurate results on the anhydrous quantity rate are desired, a favorite method is by direct weighing. Two ammonia drums, arranged in parallel, are piped in the line between the condenser and expansion valve, so that they can be filled and emptied alternately. These drums are set on platform scales, the connections to them being horizontal and sufficiently long to deflect about $\frac{1}{4}$ in. under a load at the end of about $\frac{1}{4}$ lb. The drums and contents may then be weighed without sensible error. The increase of accuracy of this procedure probably does not justify the elaboration of the apparatus.

It is useful to find the number of pounds of anhydrous per minute per ton of refrigerating effect and to compare this with the amount theoretically required as tabulated in handbooks or calculated from heat contents under prevailing conditions.

(i) Concentration and Specific Gravity of the Aqua. The concentration of aqua ammonia is determinable from its specific gravity. Given a definite solution of ammonia in water, its spe-

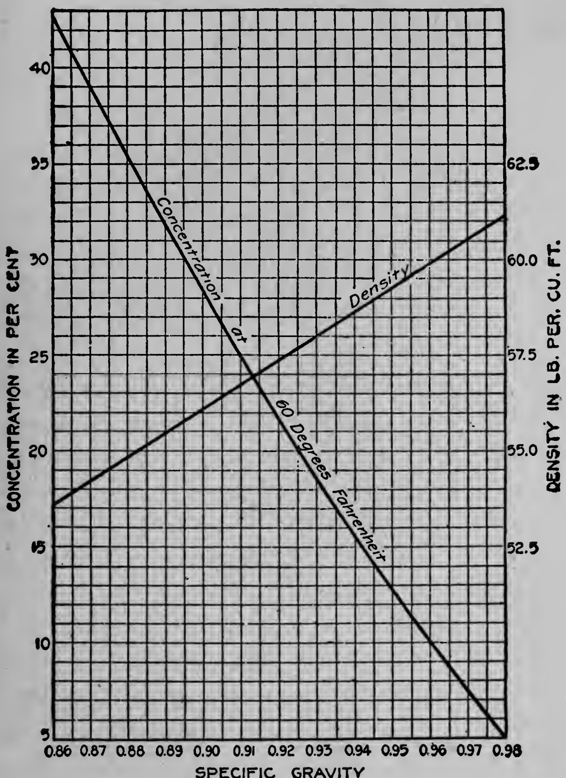


Fig. 105.—Concentration of Aqua Ammonia at 60° F., and Density Variation.

cific gravity is fixed at any one temperature, say 60°. At any higher temperature the solution occupies more volume, owing to expansion, and therefore has a lower specific gravity, although the concentration remains the same. Or to put it another way, for

any value of the specific gravity the concentration depends on the temperature.

The chart, Fig. 105, gives values of the concentration of 60° aqua, corresponding to various specific gravities.

Because of the volatility of aqua ammonia at the high temperature, and reduced pressure prevailing when a sample is drawn, considerable care must be taken when testing for concentration. The following procedure is recommended. Outlets for samples of the aqua are arranged at points where the temperatures are comparatively low; that is, on the discharge side of the pump for the strong aqua and between the absorber and weak-aqua cooler for the weak. These outlets should be fitted with short lengths of rubber tubing. A glass graduate, reading preferably in cubic centimeters, is provided, together with a hydrometer or a sensitive scales. The graduate is about half filled with water for the first trial (for the second, a somewhat different amount may be selected, depending upon the outcome of the first) and then placed in a bucket of cold brine until it and its contents are chilled to about 32°. After running a little aqua through the sampling tube, it is directed below the surface of the water in the graduate until the mixture of cold water and incoming aqua attains a temperature of about 60° as shown by a thermometer. The specific gravity of the mixture should now be taken quickly with a hydrometer or by weighing the contents of the graduate and then by calculation. Provided the temperature is within 5 to 10° of 60°, the concentration of the mixture may now be found on the chart, Fig. 105, against the determined value of its specific gravity. It is the concentration of the sample that is sought, however, and this may be calculated from the following relation:

$$\text{Conc. of sample} = \text{conc. of mixture} \div \left(1 - \frac{R}{S_m}\right),$$

in which R is the ratio of the volume of cold water, before mixing with the sample, to the volume of the mixture, and S_m represents the specific gravity of the mixture.

If scales instead of hydrometer are used to obtain the weights of cold water and mixture, then the concentration of the sample is more readily figured from the relation

$$\text{Conc. of sample} = \text{conc. of mixture} \times \frac{\text{weight of mixture}}{\text{weight of sample}}.$$

If the temperature of aqua ammonia is higher than 60°, its specific gravity will be lower by between 0.001 and 0.005 for each 10°, depending upon the concentration, the stronger solutions having a higher coefficient of expansion and therefore needing greater correction than the weaker. Under these circumstances the specific gravity S_{60} of a given solution at 60° can be figured from its specific gravity S_t at $t°$, the following formula, which gives moderately accurate results, being used:*

$$S_{60} = S_t + 0.003(t-60)(1-S_t).$$

For example, if the specific gravity is found to be 0.865 at 75°, then at 60° it equals

$$S_{60} = 0.865 + 0.003(75-60)(1-0.865) = 0.871.$$

The value 0.871 is then to be referred to the chart for the corresponding concentration.

When the reverse calculation is sought (for the density determination of aqua at high temperatures) this formula may be expressed thus:

$$S_t = \frac{S_{60} - 0.003(t-60)}{1-0.003(t-60)}.$$

(j) Calculation of Aqua and Anhydrous Weights per Minute from the Concentrations. This method is based upon the following rational relations.

* Curves presented in Marks' Mechanical Engineers' Handbook may also be used.

† See also the equation and chart given under Test 65 (*a*).

$$A_s = \frac{1 - C_w}{C_s - C_w} \times A,$$

$$A_s = \frac{1 - C_w}{1 - C_s} \times A_w,$$

$$A_w = \frac{1 - C_s}{1 - C_w} \times A_s,$$

$$A = A_s - A_w,$$

in which A_s, A_w and A are the weights in pounds per minute of the strong and weak aqua and anhydrous respectively, and C_s and C_w are the concentrations of the strong and weak aqua respectively, in pounds of NH_3 per pound of solution.

For direct measurement of these quantities, much the same methods may be employed as for the strong aqua. (See (*g*).) The reader is reminded that only one of the three quantities, A, A_w, or A_s, need be directly measured for a fairly accurate knowledge of the performance of the plant when the concentrations are known. The ammonia, whether aqua or anhydrous, may therefore be directly metered at a point selected according to the existing layout. It should be borne in mind, however, that the results depending upon concentration figures may not be exact, because of the difficulty in getting close values of the concentrations.

It is to be noted that, since these relations are between weights, a knowledge of densities is necessary to convert them into or from volumes. For the aqua this involves a measurement of specific gravity; for the anhydrous the ammonia tables may be referred to. The chart, Fig. 105, gives values of the density of aqua corresponding to those of specific gravity, the density scale being on the right.

As an example of how these relations are to be applied, let us suppose that the concentrations of the strong and weak aqua are 38 and 26 per cent respectively (the concentrations having

been found as under (*i*)), and that the number of pounds of anhydrous circulated per minute is 20. Then.

$$A_s = \frac{1-0.26}{0.38-0.26} \times 20 = 123 \text{ lbs. per minute.}$$

Referring to the chart, it is seen that 38 per cent aqua has a specific gravity of 0.873, if the aqua is at 60° F., and a density of 54.4 lbs. per cu. ft.

For the calculation of (*g*), if the temperature of the aqua at the pump is not much higher than this, then the cubic feet discharged per minute is $123 \div 54.4 = 2.26$, which is the value of V to be used in the formula for horse-power.

65. Heat Balance of a Refrigeration Plant

(Ammonia Absorption System)

Principles. The over-all heat transfers in an absorption refrigeration plant may be summarized very briefly as follows: Energy is added to the system, first by the steam supplied to the generator, second by the work of the ammonia pump, and third by the heat added to the brine during the refrigerating process. Heat is removed from the system by circulating water, first in the ammonia condenser, second in the absorber, and lastly, in the weak-aqua cooler and the rectifier. The energy added equals the heat removed, as itemized, plus or minus radiation.

A study of this over-all heat balance is useful, but before making comparisons of grand totals, it is preferable to analyze the performance of separate units. Of these it is the purpose here to deal with the generator and the absorber.

To review the action of the generator, strong aqua ammonia enters at a comparatively low temperature, to which heat is added by steam pipes. Some of the ammonia is thereby boiled out of the water holding it in solution. There result from the process a quantity of superheated anhydrous ammonia, mixed with a small

percentage of steam, and a larger quantity of weak aqua, comparatively hot, which is to be returned to the absorber. The heat in the steam supplied goes to vaporize and superheat ammonia from the form of strong aqua and to raise the temperature of the entering aqua to a higher value upon leaving. Furthermore, a certain quantity of heat is required to break the bond between the liquid ammonia and the water holding it in solution, before vaporization can take place. This quantity is additional to that required to vaporize liquid anhydrous ammonia and is referred to as the "**heat of solution.**" Experimental values of it have been made showing that it is independent of pressure and temperature and varies only with the concentration of the solution, being about 347 B.t.u. per pound for very weak solutions (approximately to zero percentage of ammonia) and diminishing as the concentration is increased to a zero value when the concentration is 60 per cent.

It is convenient to base the heat calculations of the ammonia upon a single pound of the NH_3 going through a complete cycle of temperature and state changes. From these results hourly quantities may be figured by multiplying by the number of pounds of anhydrous circulated per hour. Thus, let A represent the latter quantity and W the number of pounds of weak aqua circulated per pound of anhydrous vaporized; then the heat transferred in the generator per hour $= A \times (W \times$heat added to 1 lb. of the aqua$+$heat of solution in B.t.u. per pound of $NH_3 +$difference in "heat content" of NH_3 per pound, from tables). The quantity in the parenthesis is the heat transferred for the circulation of one pound of the anhydrous.

The product, representing B.t.u. per hour, must equal the heat given up by the steam supplied to the generator, if the losses due to radiation and steam in the superheated ammonia are ignored.

The Measurements Necessary to make for the determination of this heat balance are in part the same as those for an economy trial, for which see Test 64. In addition it is necessary to deter-

mine temperatures of the ingoing and outgoing aqua near the generator, the temperature of the ammonia vapor before it enters the rectifier and the head pressure—readings of which should be taken sufficiently to get fair averages. The vapor data are used to determine the "heat content" from the tables (see Appendix), or Mollier diagram for ammonia. For the steam-heat quantities, pressure and quality must be ascertained as well as the temperature of the condensate at the generator trap.

(a) To determine **the number of pounds of weak aqua circulated per pound of anhydrous** (that is, W), it is necessary to measure the concentrations of the weak and strong aqua (referred to as C_w and C_s, respectively) as described under Test 64 (i). The desired quantity may then be calculated from the rational formula,

$$W = \frac{A_w}{A} = \frac{1 - C_s}{C_s - C_w}.$$

A simpler procedure may be adopted through the use of the accompanying curves. Referring to Fig. 106, the lines slanting diagonally upward from left to right show values of the number of pounds of weak aqua per pound of anhydrous corresponding to various combinations of concentrations. To use these curves assume, for example, that the concentration of the strong aqua is 0.34, and of the weak, 0.24. On the chart the lines representing these values intersect between the curves marked 6.5 and 7, at about one-fifth of the distance across, so the value of W is 6.6.

This chart also gives the "heat of solution," which is plotted from the experimental values of H. Mollier. As previously remarked, this depends only upon the concentration. During the driving-off process in the generator, the concentration varies from that of the strong aqua to that of the weak. It is the average concentration, then, that determines the heat of solution. This condition is met in the chart, as before, by the intersection of the lines representing the concentrations; the location of this point

with relation to the "heat lines" (slanting downward from left to right) gives the heat of solution. For example, using the concentrations 0.34 and 0.24, the point of intersection is about four-fifths of the distance across from the 205 to the 210 B.t.u. line. Therefore the heat of solution is 209 B.t.u.

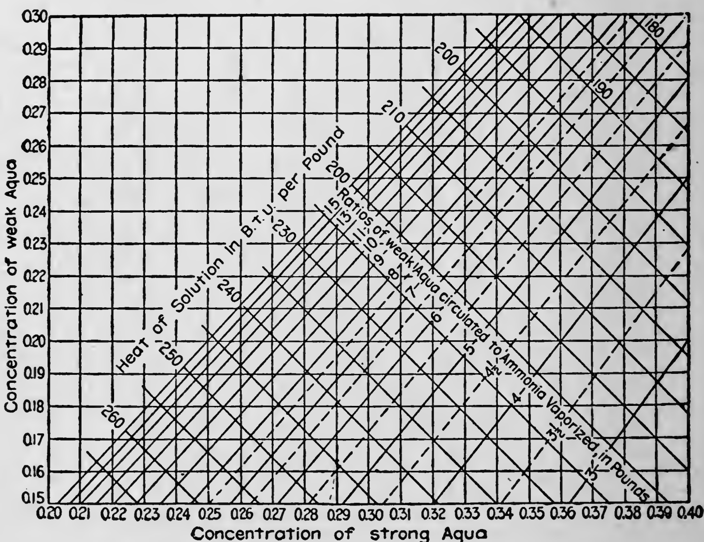


Fig. 106.—Heat of Solution of Aqua Ammonia

(b) Heat Added to NH_3 and Aqua in Generator. We now have means of finding the number of pounds of weak aqua per pound of ammonia vaporized and the heat of solution per pound. Referring to the heat equation for the generator, already given, it will be noticed that there are still to be found the heat added to one pound of the weak aqua and the difference in "heat content" of the NH_3 before and after vaporization. The latter is readily found from the ammonia tables or Mollier diagram, being the difference between the total heat of the NH_3 vapor at the head pressure and outgoing temperature and the heat of the liquid NH_3 at the temperature of the incoming strong aqua.

Considering, now, the heat added to one pound of the aqua passing through the generator, it is to be remarked that there are no tabulated values of the heat of the liquid of mixtures of ammonia and water, or of the specific heats of such mixtures. It will, however, be sufficiently accurate to assume that the heat capacity

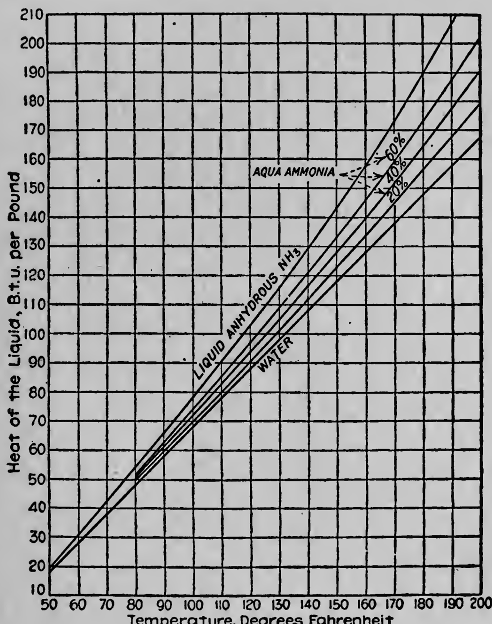


FIG. 107.—Heat of the Liquid for Water, Aqua Ammonia and Anhydrous Ammonia

of a combination of ammonia and water in definite proportion is the sum of the heat capacities of the constituents, according to their proportion. Thus, if we call the heat of the liquid of ammonia h_a and of the water, h_w (as tabulated in the ammonia and steam tables, respectively), then for a mixture of C pounds of ammonia and $1-C$ lb. of water, the heat of the liquid (aqua ammonia) = $h_aC+h_w(1-C)$. Values of this quantity at various temperatures

and concentrations are shown graphically in Fig. 107, the highest curve of which gives (vertically) the heat of the liquid of pure ammonia, the lowest curve, of unmixed water, and the intermediate curves, of mixtures of the two denoted by their concentrations. The heat of the liquid of aqua at concentrations intermediate between those shown may be obtained with sufficient accuracy by interpolation.

Coming now to detailed calculations, let us assume the following data:

Concentration, weak aqua, $C_w = 0.24$.
Concentration, strong aqua, $C_s = 0.34$.
Head pressure = 100.3 lb., gage.
Temperature of outgoing $NH_3 = 150°$ F.
Temperature weak aqua = 180° F.
Temperature strong aqua = 130° F.

From Fig. 106 is obtained the value 6.6 as the number of pounds of weak aqua leaving the generator per pound of ammonia vaporized. From Fig. 107 the heat of the liquid of one pound of 24 per cent aqua is found to be 158 B.t.u. at 180° and 102 B.t.u. at 130°. Consequently, the heat added to the aqua per pound of ammonia vaporized is $6.6 \times (158 - 102) = 370$ B.t.u. From Fig. 106, from the given concentrations, heat of solution = 209 B.t.u. From the ammonia tables, the total heat of ammonia at 100.3 lbs. gage and 150° is 609 B.t.u., and the heat of liquid ammonia at 130° (its incoming temperature) is 115 B.t.u. (also obtainable from Fig. 107). The heat added to the NH_3 to raise it from the liquid to the superheated condition is consequently $609 - 115 = 494$ B.t.u. Adding these three heat quantities ($370 + 209 + 494 = 1073$ B.t.u.), we have the heat added in the generator to effect the complete circulation through the plant of one pound of anhydrous ammonia. As before mentioned, this ignores the loss through vaporization of water with the ammonia, afterward removed by the rectifier, but this loss should be small.

(c) The Steam-heat Quantities involve the determination of the weight of steam used for a corresponding ammonia vaporization and may be made as for an economy trial. (See Test 64 (*e*)). It should be noted that

$$\text{Lb. of steam per hour} = \frac{\text{heat added to } NH_3 \text{ and aqua per hour}}{\text{heat removed from 1 lb. of steam}}.$$

(d) The Heat Transfers in the Absorber are essentially the same as in the generator, the only difference being that they proceed from the ammonia media into circulating water, instead of from steam into ammonia media, and heat is given up, instead of taken in, by the NH_3 and aqua. The aqua is pumped from the absorber at a temperature lower than that at which it entered, whereas in the generator the reverse was the case. The heat of solution is *released* in the absorber and must be removed by the circulating water, also a reverse process. The anhydrous vapor, coming from the refrigerator at comparatively low pressure, is, taken into solution by the weak aqua, consequently giving up (in addition to the heat of solution) an amount equal to the total heat of the incoming anhydrous minus the heat of the liquid NH_3 at the temperature of the outgoing (strong) aqua. It will thus be seen that the measurements and calculations for the determination of the heat given up by the absorber per pound of anhydrous ammonia are exactly the same as for the generator.

(e) Heat Removed by Condenser Water. The heat calculated under (*d*) expressed in B.t.u. per unit of time, equals the heat removed from the circulating water in the same time, plus or minus radiation. To obtain the heat removed, it is necessary to measure the circulating water for a definite time, as for an economy trial, and its temperature range between entering and leaving the absorber. (See Test 63 (*g*)).

66. * Test of a Fan Blower

Principles. There are two kinds of fan blowers called "pressure" and "volume"; the one delivering air at high pressure, the other at high velocity. The shape of the fan casing to a large extent determines its kind.

The capacity of a blower depends upon the volume of "free air" it will discharge in a given time at a given rotative speed. By free air is meant air at the pressure and temperature of the room at the time of the test. Capacity is generally expressed in cubic feet per minute.

The useful work done by a fan equals the energy imparted to the air as pressure and velocity. If W is the number of pounds of air discharged per minute, and H_1 is the pressure expressed as a head of air in feet, then WH_1 is the foot-pounds of useful work per minute represented by the pressure energy. If V is the velocity in feet per second, then $WV^2 \div 64.4$ is the foot-pounds of useful work represented by the kinetic energy. The quantity $V^2 \div 64.4$ is the head of air, in feet, equivalent to the velocity, or "velocity head," which will be referred to as H. Then the useful, or "air horse-power," is

$$\text{A.h.p.} = \frac{WH_1 + WH}{33{,}000}.$$

This is referred to by the Air Machinery Code of the A.S.M.E. as the "gross" air horse-power; the difference between it and the "net" will be pointed out under Test 67.

Selection of the Independent Variable. In the operation of a fan blower, either the rotative speed, the velocity of air, or the pressure of the air may be varied. The first is controlled according to the type of the driving engine. The air velocity or pressure may be varied at a given rotative speed only by changing the size of the outlet from the air discharge pipe. A

* See also items 22 and 44, Appendix B.

useful set of tests may be had by making the rotative speed the independent variable. The pressure is kept constant during a series of runs at different speeds; then the pressure is changed for another series, and so on. In order to keep the pressure constant when the speed is changed during each series, it is necessary to adjust the external resistance to the air. This may be done by using different nozzles or orifices as outlets from the discharge pipe. A convenient arrangement consists of a leaf shutter, similar to that of a camera, which may be readily changed to any desired diameter of outlet orifice.

A set of curves may be made between the various results and the rotative speeds. For each pressure there will be a corresponding set of curves. Taking the results from these curves on a coordinate representing one value of the speed, another set of curves may be plotted with pressure as the independent variable. A number of such sets may be made corresponding to various speeds.

Another independent variable which is sometimes used is the size of the orifice or nozzle in the outlet pipe. Pressure and velocity will then vary at each different speed, and these with the other variables may be plotted against orifice area; one set of curves for each speed of the blower.

The duration of each run need not be more than a few minutes if a velocity meter is used for air measurement. The observed quantities vary very slightly, as a rule, during each run, so that only two or three sets of observations are needed.

(a) **Determination of Horse-power Supplied.** If the fan is belt driven, a transmission dynamometer may be used to advantage. Allowance should be made for belt losses, since the horse-power supplied to the fan shaft is desired. This may be done by using the revolutions per minute of the fan shaft with the torque shown by the dynamometer to calculate horse-power, and multiplying the result by the ratio of diameters of fan pulley to dynamometer pulley. This allows for belt slip.

If the fan is driven by an electric motor, the horse-power supplied may be had from a calibration of the motor, readings of the current supplied then being necessary.

(b) Capacity. For the measurement of the air quantity a meter of the velocity type is most readily adapted. Of these, anemometers (Test 28), orifices (Test 27), and pitot tubes (Test 25) are customarily used, preference being given to pitot tubes.

Velocities may be obtained from the pitot tube either by making a complete traverse of the pipe at each determination, or by locating the velocity opening at the point of mean velocity. For details, see Test 25 (*a*). Multiplying the velocities by the cross-sectional area of the pipe gives volumes of air under the pipe conditions of pressure and temperature. To reduce these results to free air, pressures and temperatures of the air in the pipe and room must be read. In the pipe, the pressure may be read from a manometer attached to a branch from the tube leading to the static opening of the pitot meter. For room pressure, the barometer should be read.

Consider as an example the following readings. Velocity head, $h=0.5$ in. of water; pressure $=4$ ins. of water; barometer $=$ 29.9 ins. of mercury; temperature of room $=65°$ F.; temperature of air in pipe $=67°$ F. The absolute pressure of the air in the pipe is $29.9+4\div 13.6=30.2$ ins. of mercury or 30.2×0.49 $=14.8$ lbs. per square inch.

The absolute temperatures of the room and of the air in the pipe are 525° and 527°, respectively. The density of the air in the pipe may be figured from the familiar relation,

$$w=\frac{144p}{53.4T}=2.7\frac{p}{T},$$

which gives $w=2.7\times 14.8\div 527=0.0758$ lb. per cubic foot. To convert the velocity head h, in inches of water, into H, in feet of air, we have,

$$H=\frac{62.3}{0.0758}\times\frac{h}{12}=68.5h,$$

for the given data. Consequently, $H=68.5\times0.5=34.25$, and the velocity is

$$V=8.02\sqrt{H}$$

$$V=8.02\sqrt{34.25}=47 \text{ ft. per second.}$$

If the area of the pipe is 0.33 sq. ft., the cubic feet of air per minute is $60\times0.33\times47=930$. To reduce this to free air, it is multiplied by the ratio of temperatures and the inverse ratio of pressures.

$$\text{Free air per minute}=930\times\frac{525}{527}\times\frac{30.2}{29.9}=934 \text{ cu. ft.}$$

(c) **Horse-power Supplied per Thousand Cubic Feet of Free Air per Minute.** This quantity is obtained from the results of (*a*) and (*b*) being the quotient between each two corresponding values multiplied by 1000.

(d) **Air Horse-power.** For the data previously given, the weight of air delivered per minute is $930\times0.0758=70.5$ lbs. The velocity head is 34.25 ft. of air, and the pressure head, $68.5\times4=274$ feet of air. Then the air horse-power is

$$\text{A.h.p.}=\frac{70.5(34.25+274)}{33{,}000}=0.659.$$

(e) **The Efficiency** equals the air horse-power divided by the horse-power supplied.

Problem 66_1. In the test of a blower, if gasoline, specific gravity $=0.75$, is used as a gaging fluid with a pitot tube, what is the velocity head in feet of air, if $h=4.5$? Pressure is 6 ins. of the same fluid. Barometer is 30.3 ins. mercury. Temperature is 70° F. *Ans.*, 229 ft.

Problem 66_2. In the preceding how many foot-pounds of work will be done in one minute if pipe dia. $=6$ in.? *Ans.*, 58100.

Problem 66$_3$. In the operation of a blower, the volume discharged varies very nearly as the rotative speed. The pressure is due to centrifugal force. From these facts, deduce how the power will vary with the speed.

67. * Test of a Reciprocating Air Compressor

Principles. In the operation of an air compressor, it is the purpose to increase the pressure of the air supplied so as to make available the energy it contains. As this energy is to be used when the air is cool, it is desirable to compress the air isothermally. If it is allowed to heat, the pressures during compression are higher and more work is required for compression. For this reason, water jackets and intercoolers are used, the heat removed by them being a saving.

Referring to Fig. 108, the dotted lines show an ideal air compressor diagram from a cylinder without clearance in which the air is discharged at a pressure, 4_2-5_1, equal to that in the delivery main. The supply is drawn in at atmospheric pressure along the line 6_1-2_1 and then compressed isothermally along 2_1-4_2.

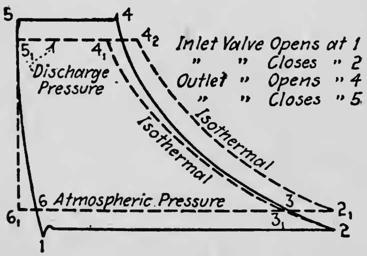


Fig. 108.—Ideal and Actual Air Compressor Diagrams.

The actual indicator diagram varies from the ideal one as shown by the full lines, 1–2–4–5. The friction resisting the motion of the air through inlet and outlet valves and ports necessitates a lesser pressure than atmospheric to draw in the air, and a greater pressure than that in the delivery main to discharge it. Consequently, work is lost as represented by the areas $4-5_1-5$ and 1–2–3–6. The actual compression line, 2–4, must be above isothermal because of the impossibility of perfect cooling, and this results in the loss rep-

* See also Appendix B, items 22 and 44.

resented by the area between the lines 2–4 and 2–4_1. These are "compression" losses.

When the outlet valve closes at point 5, an amount of air equal to the clearance volume is entrained in the cylinder. At the beginning of a new cycle, this air expands so that the effective stroke for drawing in air begins at point 6 instead of 6_1. This point occurs later, the higher is the delivery pressure (at 5). When the inlet valve closes at 2, the air in the cylinder is rarified because of the suction, and it is not until 3 is reached that its pressure becomes atmospheric. The effective stroke for drawing in air is therefore represented by line 6–3 instead of the full length of the diagram. This is a "volumetric" loss, since less air is delivered than would be if the full stroke were effective. Another volumetric loss, not appearing in the diagram, is due to leaky valves and pistons, similar to slip of a pump.

After the air in the delivery pipe has been cooled down to room temperature, it contains energy, due to its pressure above atmospheric, available for performing work. If this air could be expanded isothermally to its original pressure, there could be regained all of the energy necessary to compress it isothermally. This, however, cannot be done, the actual expansion being more nearly adiabatic. There is a final loss, then, due to the lack of availability of all the energy added to the air.

These various losses are indicated in certain expressions for efficiency, to be defined in the following, which are generally figured as air compressor test results. It is first necessary to define:

Gross Air Horse-power. This is the same as the indicated horse-power of a steam engine, being figured from the mean effective pressures of the indicator diagrams of the air cylinders.

Net Air Horse-power. This is the horse-power required to compress isothermally from room pressure and temperature to the pressure in the delivery main, a mass of air equal to that actually delivered.

The Mechanical Efficiency equals the gross air horse-power divided by the horse-power supplied to the compressor.

The Volumetric Efficiency is the cubic feet of free air actually delivered in a given time divided by the low pressure piston displacement in the same time.

The Efficiency of Compression is the net air horse-power divided by the gross air horse-power. This efficiency covers all the losses of the air cylinders due to faulty valve action, leakage, lack of cooling, etc., except that due to mechanical friction.

Over-all Efficiency is the net air horse-power divided by the horse-power available for running the compressor.

(a) **Capacity** may be expressed as the number of cubic feet of air discharged per minute at the pressure in the delivery main corrected to room temperature, or as the number of cubic feet discharged per minute reduced to free air, that is, air of the same temperature and pressure as that supplied to the compressor.

Let Q = cubic feet of air per minute discharged, as found by meter;
T = absolute temperature, degrees Fahrenheit of air discharged;
P = absolute pressure, pounds per square inch, air discharged;
t = absolute temperature, degrees Fahrenheit, of air supplied (room);
p = absolute pressure, pounds per square inch, air supplied.

Then the capacity in terms of compressed air is

$$Q_1 = Q \times \frac{t}{T},$$

and in terms of free air is

$$Q_2 = Q \times \frac{t}{T} \times \frac{P}{p} = Q_1 \times \frac{P}{p}.$$

T and t are to be determined by thermometers, one in the room near the air inlet to the compressor; the other in the discharge pipe just beyond the last cylinder or receiver. p is obtained by a barometer and reduced to pounds per square inch; P, by a pressure gage set close to the thermometer measuring T.

The cubic feet of air Q flowing in the delivery main is a quantity difficult to measure accurately. The methods outlined under Part I are applicable, but, owing to the high pressures and large volumes, are not altogether satisfactory. Gasometers give the most reliable results. Two tanks similar to Fig. 56 may be connected together at the bottoms by a water pipe, and air piping arranged so that they may be alternately filled with the discharge from the compressor. As air enters one, water is displaced into the other until a mark on the gage glass is reached. The air is then diverted into the other tank, the water returning to the first one which at the same time discharges the air it has just been filled with. The cross-sectional area of the tanks and the distance between upper and lower water levels, together with the number of fillings per minute, give the quantity sought. It is a good plan to pipe the air through a four-way cock so arranged that one motion connects one tank with the air delivery line and the other tank with the atmosphere.

(b) **Mechanical Efficiency.** The gross air horse-power is figured from the indicator diagrams from the air cylinders. The horse-power supplied to the air compressor, if it is steam driven, is the indicated horse-power of the steam cylinders. If it is belted or geared to the source of power, the horse-power supplied is that received at the compressor shaft, and should be obtained as for a fan blower, Test 63 (*a*). From these results the mechanical efficiency may be obtained.

(c) **Volumetric Efficiency.** The length of the line 6–3, Fig. 108, divided by the length of the diagram 6_1–2_1, is often referred to as the "apparent" volumetric efficiency; apparent, because

it does not take account of slip and leakage. The "true" volumetric efficiency is,

$$Q_2 \div L(A_h + A_p)N$$

in which L is the length of the stroke in feet, A_h and A_p are the areas in square feet of the piston at the head and power ends, respectively, and N is the number of double strokes per minute. If the compressor is single-acting, A_p should be omitted.

If the compressor has more than one cylinder, the expression applies to the low-pressure cylinder.

(d) **Efficiency of Compression.** The net air horse-power must first be figured. The work done during isothermal compression of a gas is mathematically equal to

$$\text{abs. pressure} \times \text{volume} \times \text{hyperbolic log.} \frac{\text{final abs. pressure}}{\text{initial abs. pressure}}.$$

The volume to be considered in this case is that accounted for in the delivery main, corrected to room temperature. Then, using the notation given under (*a*),

$$\text{Net A.h.p.} = \frac{144PQ_1 \log_e \dfrac{P}{p}}{33{,}000}.$$

Dividing this by the gross air horse-power gives the required result.

(e) **Over-all Efficiency.** The horse-power available for running the compressor, in the case of a belted or similar machine, is the horse-power supplied to the shaft, the same as the denominator of the fraction expressing mechanical efficiency. For a steam driven machine, it is the horse-power available in the steam supplied. This may be taken as the horse-power that would be developed if the steam worked on the ideal Clausius cycle.

If E_c is the Clausius cylinder efficiency, figured according to Test 45 (c), then

H.p. available in steam = I.h.p. in steam cylinder $\div E_c$.

Dividing the available horse-power into the net A.h.p. gives the over-all efficiency.

(f) Separation of Losses. A precise separation of all the losses cannot readily be made for the reason that they merge into each other. For example, mechanical friction in the air cylinder adds heat to the air which makes the loss due to ineffective cooling greater. If there is leakage of air past the inlet valve during the compression, the compression curve will approach more nearly to the isothermal, making an apparent gain in the jacket's efficiency, but an actual loss as far as power and capacity are concerned. Again, if the inlet valve is mechanically operated and opens too early, the compressed air in the clearance space escapes and the apparent volumetric efficiency is improved, but power is lost.

If it is borne in mind that the losses as shown by the indicator diagram are not precise, an idea may be formed of them by an analysis of a representative diagram from a test. Fig. 109 is such a one. Through 2_1, vertically over 2, a horizontal should be drawn to represent atmospheric pressure. Through 2_1 an isothermal is drawn referred to a volume axis through 5. Through 2, another isothermal is drawn referred to the volume axis OY, located from the left of the diagram at a distance proportional to the clearance of the cylinder. Finally, a horizontal through

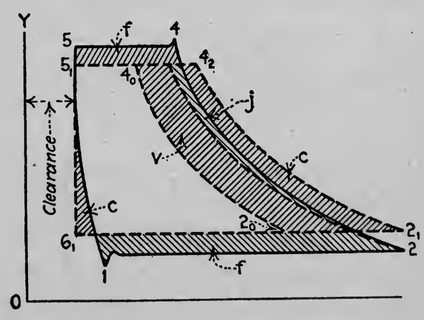


Fig. 109.—Air Compressor Losses.

4_2 is laid off to represent the pressure in the delivery main. The ideal diagram is then 6_1–2_1–4_2–5_1.

The loss due to fluid friction at the valves and ports is represented by areas *f*, *f*.

The loss due to insufficient cooling by the jacket water is represented by area *j*.

The loss in capacity due to volumetric inefficiency other than leakage is represented by the areas *c*, *c*. This is not a loss of power supplied, since the work is not done. It is merely a diminution of the capacity of the compressor to deliver net horse-power.

The loss in capacity due to leakage is also a loss of power, because work is done on the air which leaks out of the system just as it is done on that which is discharged into the delivery main. To estimate this loss, a point 2_0, Fig. 109, is located so that the distance 6_1–2_0 = distance 6_1–2_1 × true volumetric efficiency. Through 2_0 an isothermal is drawn referred to 5–6_1 as the volume axis. The diagram 6_1–2_0–4_0–5_1 then represents the net horse-power, and the area *v*, the loss of power due to leakage.

These areas may be integrated and expressed as per cents of the area representing the gross horse-power.

If the compressor has more than one cylinder, the indicator diagrams should be combined by reconstructing them on the same scale. The line 4_2–5_1 will then represent the receiver pressure. The next cylinder diagram will have this line as a datum in the same way that the low-pressure cylinder diagram has the atmospheric line as a datum. The general method outlined in Test 46 (*f*) may be used for construction.

(g) **Heat Measurements.** The heat equivalent of the work delivered to the air piston

= heat added to the jacket water
+ heat in the air in delivery main, above room temperature
+ heat radiated.

This follows from the fact that no energy is added to air when it is compressed, or expanded, isothermally, the isothermal being a "constant energy" condition. The heat removed per minute by the cooling water may be measured by taking weight and thermometer readings. The heat in the air may be figured from the data for the other results and from the specific heat of air. These quantities, considered with regard to the other results, will give a more complete idea of the efficiency of cooling. They may be expressed in horse-power units by dividing by the number of heat units per minute equivalent to one horse-power, 42.42.

Problem 67_1. An air compressor delivers 8.3 cu. ft. of air per minute, at 100 lbs. pressure absolute, and 220° F. What is its capacity in cubic feet of free air per minute; temperature of the room being 75°, and pressure, 29.5 ins. of mercury? What is its capacity in cubic feet of compressed air corrected to room temperature? *Ans.*, 45.1 and 6.52 cu. ft.

Problem 67_2. The compressor of the preceding problem was two-stage, 14-in. and 9-in. bore by 12-in. stroke, and 100 working strokes per min. per cylinder. What is the volumetric efficiency from the same data? *Ans.*, 42.2%.

Problem 67_3. What is the net air horse-power from data of Problem 67_1? What is the horse-power available in the compressed, cooled air? *Ans.*, 5.5 H.p.

Problem 67_4. How can the area *j*, Fig. 109, be roughly figured from heat measurements?

68. Test of a Hydraulic Turbine

Principles. The horse-power delivered by a hydraulic turbine is as indicated by a dynamometer applied to its shaft. The horse-power available is that of the water which drives it, and is proportional to the difference in level between the head race and tail race, and the weight of water flowing per minute. The quotient between these horse-powers equals the over-all efficiency of the plant.

When the turbine is supplied from a pipe line under pressure, as in the arrangement of wheels of the Pelton type without a

draft tube, the horse-power available to the machine may be taken as that due to the pressure and velocity energy of the water just before entering the turbine. The machine is then not charged with the friction losses of the pipe line or with the losses due to its position in relation to the tail race.

Selection of the Independent Variable. In operation, any or all of the following may be varied. First, gate or needle valve opening; second, rotative speed; and third, brake horse-power. For test purposes, the head on the turbine may also be varied.

Change of the gate opening varies the amount of water supplied. Change of the brake horse-power is accompanied by a change of speed if the gate opening is left the same.

Laboratory tests are usually made at a constant head. The gate is adjusted at a predetermined opening for one series of tests. Then, by regulating the brake, the speed is varied for this series, and data obtained at each speed. Another series of tests is then made at a different gate opening, and so on until the full range has been covered.

If it is desired to test the performance at different heads, all or part of the above tests may be repeated after changing the head.

It is useful to plot brake horse-power and efficiency at each gate opening against speed.

The duration of each run need be only long enough to obtain accurate measurement of the rate of water supplied when the head remains practically constant, once uniformity of conditions has been established.

(a) Available Horse-power. This equals

$$\frac{WH}{33,000} \quad \text{or} \quad \frac{W(2.3p+V^2\div 2g)}{33,000},$$

in which W is the weight of water per minute in pounds; H, the difference in level in feet, for the first case; and for the

second, p is the pressure in the water pipe in pounds per square inch; V, the velocity in feet per second, and g the acceleration of gravity.

It should be noted that the first expression is the energy available to the turbine *plant*, the second is the energy available to the *turbine*.

The most convenient method of measuring the water supplied is by weirs, the quantity usually being large. For small turbines, calibrated tanks may be used instead.

When the difference of level between the head race and tail race is measured, two datum marks may be made at the approximate positions of the upper and lower levels. The variations of the levels from these marks may be noted at a number of time intervals during each run. From these data, the average head available to the turbine plant may be figured, the vertical distance between the marks having previously been measured.

When the second relation for available horse-power is used, the pressure may be obtained by a pressure gage just back of the control valve. The velocity is readily obtained from the previously measured water quantity, the cross-sectional area of the pipe being known. Sometimes the velocity energy is small enough to be negligible.

(b) **Hydraulic Efficiency** is obtained by dividing the horse-power available to the turbine into the brake horse-power. The latter may be measured by one of the forms of friction brake.

(c) **Determination of Best Operating Speed.** The curve of efficiency vs. speed is something like an inverted U. The speed at the highest point of this curve gives maximum efficiency. It is approximately that speed which gives a peripheral velocity of one-half that of the jet for impulse wheels. For other types, it depends upon the characteristics of the turbines.

Under ideal conditions, the water upon leaving the turbine

vanes would have no velocity except that due to the acceleration of gravity, since velocity of the off-flow means lost energy. With some types of turbine the best speed may be ascertained by watching the off-flow and determining that speed which is accompanied by the most nearly vertical descent of the water as observed by the eye.

Problem 68_1. What is the pressure energy in foot-pounds per minute available to a Pelton wheel supplied with 10 cu. ft. of water per minute at 100 lbs. pressure? *Ans.*, 144,000 ft.-lbs.

Problem 68_2. In the preceding problem, if the pipe supplying the water is 2 ins. inside diameter, what is the velocity energy? What is the total horse-power? *Ans.*, 565 ft.-lbs. per min.; 4.38 H.p.

69. Test of a Hydraulic Ram

Principles. The hydraulic ram is a pump which uses the energy of a large volume of water under a low head in order to deliver a part of the water at an increased head. This is accomplished by establishing a flow through a waste valve at the foot of the supply pipe. When the velocity thus started attains a certain value, it closes the waste valve, whereupon the kinetic energy of the previously moving column in the supply pipe is converted into pressure great enough to open a valve into the discharge pipe. Water is then delivered at the increased pressure until the pressure energy is reduced to the static condition, when the discharge valve closes, the waste valve opens, and the cycle recommences. Fig. 110 shows diagrammatically the arrangement of piping of a hydraulic ram.

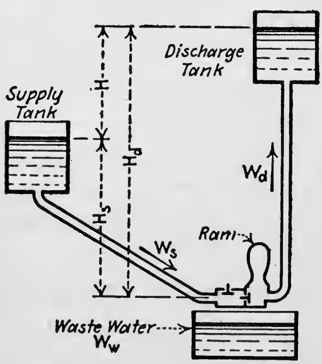


Fig. 110.—Hydraulic Ram.

Two values for the useful work of a ram may be obtained, depending upon the level from which the head pumped against is dated. Referred to the level of the supply water, the useful work is that corresponding to the elevation of a weight of water through the distance H, Fig. 110. Referred to the level of the ram, it is the work performed in lifting the same weight of water through the distance H_d. The one result leads to what is known as Rankine's efficiency, the other to D'Aubisson's. No confusion should arise through the existence of the two different efficiencies. Whether the one or the other should be used depends upon whether one is interested in pumping water from the supply level or from the level of the ram.

Selection of the Independent Variable. Laboratory tests are generally conducted under a constant supply head, although this may be varied in the same way as for the test of a water turbine. If the supply head is maintained constant, either the number of strokes per minute of the waste valve or the discharge head may be made the independent variable.

(a) **Capacity** is expressed in gallons of water pumped per twenty-four hours. This may be measured by catching the discharge in a pail, or larger vessel if necessary, during a counted time. Instead of pumping against a static head as in regular operation, the discharge may be throttled by means of a valve in the discharge pipe which may then be short enough to collect conveniently the water delivered. The desired head is ascertained and regulated by the aid of a pressure gage between the ram and the throttle valve.

If the number of strokes per minute of the waste valve is the independent variable, it can be varied by changing the lift of the valve, or adjusting the spring tension if it is operated by a spring.

(b) **Efficiencies.** Let W and H stand for the weight of water flowing per minute in pounds and head in feet, respectively; and let the subscripts s, w, and d refer to the supply, waste,

and discharge, respectively, as indicated by Fig. 110. According to Rankine's efficiency a weight of water W_w must fall H_s ft. in order to raise W_d lbs. of water H ft. Therefore,

$$\text{Rankine's efficiency} = \frac{W_d H}{W_w H_s}.$$

According to D'Aubisson's efficiency, the energy available to the ram is the denominator of the following expression, and this energy accomplishes $W_d\ H_d$ ft-lbs. of work.

$$\text{D'Aubisson's efficiency} = \frac{W_d H_d}{W_s H_s}.$$

Since W_d was measured for the capacity determination, it is necessary only to measure the waste water to determine W_s. This may be done by allowing the waste to collect in a calibrated tank. The head H may be found from the pressure gage readings, and H_s measured before the test. It is a good plan to use a float valve in the line which feeds the supply tank so that a constant level may be maintained in it automatically.

(c) **Curves** of capacity, total amount of water supplied in gallons per twenty-four hours, and efficiency against the independent variable may be plotted from the data previously mentioned.

Problem 69_1. The capacity of a ram operating against 20 lbs. pressure is 1042 gals. per twenty-four hours, and its efficiency (Rankine) is 35 per cent. The supply level is 10.5 feet above the ram. How much water passes through the waste valve in gallons per twenty-four hours?

Ans., 10,100 gals.

Problem 69_2. Using the data of the preceding problem, what is the D'Aubisson efficiency? *Ans.*, 41%.

70. Test of a Centrifugal Pump

Principles. The useful work of a centrifugal pump is the product of a weight of water delivered in a given time in pounds

and the total head pumped against in feet. The total head includes the friction heads of suction and discharge pipes and the velocity head.

The work supplied to the pump is that received by its shaft in the case of a belt-driven machine, or one directly connected to an electric motor. In the case when the drive is a directly connected steam engine, both the drive and the pump are generally tested as a single unit as would be a reciprocating pump.

Selection of the Independent Variable. The rotative speed, quantity discharged, or head pumped against may be varied independently for a test. When one is varied arbitrarily, another is kept constant, and the third becomes a dependent variable. Two series of tests are useful; one at constant speed and the other at constant head, the independent variable being head and speed, respectively. When the speed is kept constant, the head may be varied by throttling the discharge, and when the head is kept constant, the speed is varied by control of the motor driving the pump. In the latter case, each change of speed must be accompanied by a change in the valve opening in the discharge pipe in order to maintain a constant head.

(a) Capacity may be expressed in gallons per twenty-four hours or per minute. The usual methods of measuring water rates mentioned under the heads of hydraulic turbines and reciprocating pumps, may be used.

(b) Horse-power Supplied. The method of measuring this item depends upon the drive. If belt-driven, a transmission dynamometer is applicable. If motor driven, the motor should be calibrated (Test 74). If driven by a steam engine or turbine, it is generally more convenient to find the horse-power supplied to the set rather than to the pump alone.

(c) The Water Horse-power may be calculated from the relation

$$\text{W.h.p.} = \frac{WD}{33{,}000}.$$

W is the weight of the water discharged, in pounds per minute, measured as under (*a*). D is the total head pumped against, in feet, and may be determined as follows. Connect a pressure gage to the discharge pipe as close to the pump as possible. To the suction pipe, also close to the pump, connect a Bourdon gage or a mercury manometer. Call the indications of these instruments

P_d = discharge gage reading, pounds per square inch;
p_s, P_s = suction gage reading, inches of mercury or pounds per square inch;
h = manometer reading, inches of mercury.

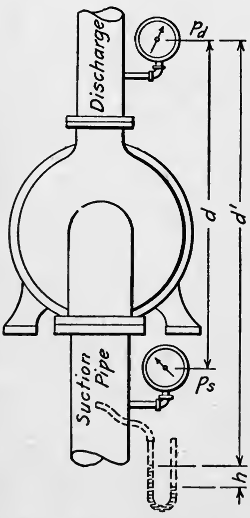


FIG. 111.—Measurement of Head.

Also let d and d' represent the distances in feet between the gages as shown by Fig. 111.

Assuming the diameters of the discharge and suction pipes equal, the velocity heads in these pipes will be equal. Then, since the *energy WD added* by the pump to the water equals the *difference in energies* on the discharge and suction sides (note that p_s is a negative pressure, counting from atmosphere),

$$WD = W \times 2.3P_d + d - W \times (-1.13p_s),$$

and

$$D = 2.3P_d + d + 1.13p_s, \quad . \quad . \quad . \quad . \quad (1)$$

which relation is to be used when the suction pipe pressure is read with a vacuum gage. Similarly,

$$D = 2.3P_d + d' + 1.13h, \quad . \quad . \quad . \quad . \quad (2)$$

which is the appropriate relation when a mercury manometer is employed.

If the level of the water supplied is higher than the pump, a pressure gage is used in the suction pipe instead of a vacuum gage. Then the term $1.13p_s$ in equation (1) changes to $-2.3P_s$, P_s being in pounds per square inch. Or, if a manometer is used, the term $1.13h$ in Eq. (2) changes to $-1.13h$.

If the diameters of discharge and suction pipes are different, the change in velocity head should be credited to the pump. If V_d and V_s stand for the velocities in feet per second on the discharge and suction sides, respectively, then there should be added to D (as obtained from Eq. (1) or (2)), the following:

$$\frac{V_d^2 - V_s^2}{64.4}.$$

The method of finding the head when testing a reciprocating pump (see p. 254), may also be used, but this does not credit the pump with the friction head in the suction pipe.

(d) **Hydraulic Efficiency** may be calculated by dividing each value of water horse-power by the corresponding value of the horse-power supplied.

(e) **Curves** may be plotted, from the data obtained, as follows: For constant speed tests, power supplied, capacity, and efficiency against head. For constant head tests, the same results plotted against speed. Sometimes speed or head, power supplied, and efficiency are plotted against quantity discharged.

Problem 70_1. How would you determine separately the work done by the various impellers of a stage centrifugal pump?

Problem 70_2. Does or does not the measurement of total head by gages, outlined under (c), include the friction head of suction and discharge pipes, and why? Does it include the friction head of the water in passing through the pump, and ought this head to be included? Why?

71. Test of a "Power" Pump

Principles. By "power" pump is meant a reciprocating pump driven by a crank shaft which receives its power through a

belted pulley, or gears from a motor. The general principles to be studied for testing are similar to these previously outlined under Test 50.

(a) **Water Horse-power.** See Tests 50 (*a*) and 70 (*c*).

(b) **Available Horse-power.** This may be found as for a centrifugal pump, Test 70 (*b*), when the drive is by belt or geared motor.

(c) **Mechanical and Fluid Losses and Efficiencies.** The difference between the water horse-power and the indicated horse-power equals the hydraulic losses. The difference between the indicated horse-power and item (*b*) equals the mechanical losses. Expressions for the corresponding efficiencies are obvious. The total efficiency is item (*a*) divided by item (*b*).

(d) **Slip and Capacity.** See Test 50 (*d*) and (*e*).

72. Test of a Hoist

Principles. A hoist is a machine by which a large weight may be lifted by the application of a small force. This is accomplished usually by passing a chain, or chains, over a number of wheels geared together in such a way that a large motion of one end of the chain downward produces a small motion of the other end upward, whereby a mechanical advantage is obtained.

The Ideal Mechanical Advantage is the ratio of a distance moved through by the driving chain to the corresponding distance moved through by the following chain. Referring to Fig. 112, representing a hoist diagrammatically, this ratio is D/d. If there were no friction, this would be the ratio of the weight lifted, W, to the force applied F. As there is friction,

The Actual Mechanical Advantage equals the ratio of the weight lifted to the force applied, or W/F.

The efficiency of a hoist equals the work done in lifting the weight through any distance divided by the work done by the applied force acting through the corresponding distance.

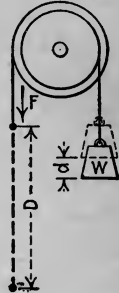


Fig. 112. Hoist.

If the efficiency of a hoist is more than 50 per cent, the weight lifted will return by gravity when the hoisting force is removed unless there is a locking device. Where this is provided, it is arranged to lock against the force of gravity only, and not against a force applied to the driving chain for the purpose of lowering the weight.

(a) The Ideal Mechanical Advantage may be determined by actual measurement of the distances, D and d, Fig. 112. As d is generally very small compared with D, the result by this method may not have the desired degree of accuracy. A better method consists of counting the numbers of teeth of the various gears of the hoist and figuring from thedata obtained, by the principles of kinematics, the velocity ratio of driver to follower.

(b) The Actual Mechanical Advantage may be determined by measuring the force applied to lift various loads. The desired results vary with the load, and may be plotted against its values. The loads may be applied by using various dead weights, sufficient to cover the working range of the hoist. The force required to lift these weights may be measured by applying the force through a spring balance hooked into the driving chain. Owing to the difficulty of reading a moving instrument, this is not a very accurate method. A better one is for the experimenter to stand on a platform scales and to apply the driving force at a uniform rate with one hand and at the same time, with the other hand, to balance the scales by adjusting the jockey weight. The weight of the experimenter minus the reading of the scales then equals the force applied to the hoist.

Several determinations of the force applied should be noted at each load, and their mean used to obtain the actual mechanical advantage at that load. It should not be attempted to make mental averages.

(c) **Efficiency** equals (see Fig. 112)

$$\frac{\text{Work done}}{\text{Work applied}}=\frac{W\times d}{F\times D}=\frac{\frac{W}{F}}{\frac{D}{d}}=\frac{\text{Actual mechanical advantage}}{\text{Ideal mechanical advantage}}.$$

Consequently each result under (*b*) may be divided by that under (*a*) to get the corresponding efficiency. The efficiencies should be plotted against the loads.

Problem 72_1. Assuming that the friction losses of a hoist are constant throughout its working range, deduce the form of, and sketch roughly, curves between force applied and load lifted and between efficiency and load lifted. Will these curves pass through the origin or not, and why?

Problem 72_2. A differential hoist has 12 chain notches on the driving wheel and eleven on the smaller. What is its ideal mechanical advantage? *Ans.*, 24.

Problem 72_3. The efficiency of a hoist is 60 per cent and its ideal mechanical advantage is 40. What load will a force of 4 lbs. lift? *Ans.*, 96 lbs.

Problem 72_4. Prove that a hoist having an efficiency greater than 50 per cent needs a locking device.

73. Tests of Lubricating Oils

Principles. The most indicative results of the relative values of lubricating oils are obtained from tests of viscosity, flash and chill points, and coefficient of friction.

Viscosity manifests itself as internal friction of the oil, a property which opposes the motion of the particles upon themselves, resulting in a reluctance to flow. It is related to, but is not proportional to the density. Since it opposes motion, it is an undesirable property, but there is such a thing as too little vis-

cosity for good results. If the oil flows too readily, it may be squeezed out of the space between the surfaces it should lubricate, and the total amount of oil needed for proper lubrication might then become excessive. This depends largely upon the pressure between the bearing surfaces.

The term "body" is also used to denote viscosity.

The flash point is the temperature at which the oil will vaporize fast enough to form an explosive mixture with air. This temperature should be higher than the working temperature to be encountered in the service of the oil. Consequently it should be judged in connection with its service.

The burning point is the temperature at which a body of the oil considered will burn when a flame is placed a short distance over its surface.

The chill point is the temperature at which the oil loses its fluidity. Its determination is necessary only when the temperatures during service are low.

For more complete details concerning the requirements for and usual constants of lubricating oils, see Stillman's "Engineering Chemistry" or Marks' "Mechanical Engineer's Handbook."

(a) **The Specific Gravity** is of interest in connection with viscosity, since a high value of the one generally indicates a high value of the other. It may be determined by balancing a column of oil in one leg of a U-tube against a column of water in the other leg. Then the specific gravity of the oil equals the height of the water column divided by the height of the oil column. A more convenient method is to use a hydrometer if one is available.

Hydrometers are obtainable graduated to indicate specific gravity relative to the weight of water, but more usually they are graduated according to an arbitrary scale. The so-called Baumé scale is largely used in the United States, although the more logical scale based on the weight of water is preferable.

Hydrometers so graduated indicate Baumé degrees (deg. Bé.). The relation between these degrees and specific gravity on the water basis is, for liquids heavier than water,

$$\text{Specific gravity} = 145 \div (145 - \text{deg. Bé.})$$

and for liquids lighter than water,

$$\text{Specific gravity} = 140 \div (130 + \text{deg. Bé.})$$

(b) **Viscosity.** This is a purely relative measurement. There are different standards for its expression, but, in general, it may be taken as the ratio of the time required by a given volume of the oil considered to flow through an orifice of a given size under a given head, or drop in head, to the time required by the same volume of another liquid, as water, to flow under the same conditions.

The temperature of the oil has to do with its viscosity, so for comparative results the temperature must be kept constant during all tests. It should be remembered that the temperature during a test may be quite different from that in actual service, so the results are strictly comparative and not always indicative of the true merits of oils to be used at high temperatures.

Apparatus for viscosity tests, or "viscosimeters," may be readily improvised. It is desirable that the vessel containing the oil be water-jacketed to maintain a constant temperature. The orifice should be about $\frac{1}{16}$ in. in diameter. The head on the orifice may start at about 6 ins. and end at about 4 ins., the total volume of oil passing through the orifice being that between the levels at the start and finish. Some forms of viscosimeters are arranged so that the flow shall be under a constant head.

(c) **Flash, Burning, and Chill Points.** The flash point may be found by placing the bulb of a mercury thermometer in a

porcelain dish filled with oil to be tested, and heating the oil over a Bunsen burner until a flash may be obtained from it by passing a lighted taper over its surface. The temperature of the oil when this happens is noted as the flash point. The burning point is found similarly, by carrying the heating to a point at which the flash causes a burning of the oil in the dish. The chill point may be found by chilling a small amount of the oil in the bottom of a test tube by surrounding it with a freezing mixture until it congeals. The test tube is then removed and the oil stirred with a thermometer until it has warmed sufficiently to flow from one end of the tube to the other. The temperature is then noted as the chill point.

(d) **Coefficient of Friction.** To determine this, an oil testing machine should be used. See Test 35. The oil to be tested should be used under the same bearing pressure and the same temperature as it will meet in service, as far as is possible. Comparative tests of different oils should be made under uniform conditions of temperature, pressure and velocity, or else the results will not be comparable.

(e) **Endurance Tests** have to do with the total amount of oil necessary to secure required lubrication. An oil that is satisfactory in all other respects may be of prohibitive cost because its lack of body may necessitate a large rate of feed. An idea of the endurance of an oil may be formed by supplying the bearing of an oil tester with a limited amount of it and noting the time required to raise the temperature of the bearing a predetermined amount. Another method is to note the time required to raise the coefficient of friction a predetermined amount. Still another is to measure the amount of oil during a given period of time when it is fed at a rate just sufficient to prevent a rise of temperature.

Problem 73_1. If the specific gravity of an oil is 0.85, what is its specific gravity in degrees Baumé? If specific gravity is 1.15, what is its value in degrees Baumé? *Ans.*, 34.8°, 18.9°.

74. Horse-power Test of an Electric Motor

Principles. In mechanical engineering tests, it is often desirable to know the horse-power delivered by an electric motor to the unit it drives, as, for example, a motor-driven centrifugal pump or blower. It then becomes necessary to make a separate test of the motor. This can be done, of course, with a Prony brake by getting corresponding values of B.h.p. and current supplied. Such procedure is not always convenient, however, nor are the results as accurate as when the useful horse-power is found by the measurement of losses. Consequently, the latter method will be described. The following notation will be used.

V = voltage of line;
A = armature current in amperes, motor loaded;
A_0 = armature current, motor running free;
F = field current, amperes;
R = resistance of armature, ohms.

Now the power delivered at any load, in watts, is

$$746 \times \text{B.h.p.} = \text{watts input} - \text{losses}.$$

The losses may be grouped as constant and variable. The **variable loss** to be considered is due to the armature resistance, increasing as the armature current increases, and equals A^2R.

The **constant losses,** entering the calculation, are due, first to friction and windage, and second, to hysteresis and eddy currents (the so-called " iron losses "). These losses may be considered constant at all loads, provided only that the speed is approximately constant. Now, when the motor is operated without load, the watts input equals the losses; and, since A^2R losses at no load are very small, it follows:

$$\text{Constant losses} = V(A_0 + F).$$

The watts input, under load, obviously equals

$$V(A+F).$$

Substituting these values of input and losses in the equation for B.h.p., we have

$$\begin{aligned} 746 \text{ B.h.p.} &= V(A+F) - A^2R - V(A_0+F) \\ &= VA + VF - A^2R - VA_0 - VF \\ &= A(V-AR) - VA_0; \end{aligned}$$

and

$$\text{B.h.p.} = .00134\{A(V-AR) - VA_0\}.$$

(a) Determination of Horse-power Output. In the equation last given, VA_0 and R are taken as constant, and may be predetermined. Then, at any load applied to the motor, it is only necessary to measure the armature current, A, corresponding to that load, and the voltage; V, in order to calculate the B.h.p.

The armature resistance, R, may be determined by the "drop of potential" method. The armature being stationary, connect the armature leads to the line with a *resistance* and ammeter in series, being *careful to avoid injury to the coils with excessive current.* Measure the amperes flowing, and the E.m.f. drop across the brushes with a voltmeter. Then

$$\text{Resistance, ohms} = R = \frac{\text{Voltage drop}}{\text{Amperes}}.$$

This should be repeated for a number of different positions of the armature, and the average resistance from the various positions taken. Repeat, also, for another value of the current.

Next, the motor should be connected as in service, but with an ammeter in the armature circuit, and a voltmeter to indicate the voltage at the armature terminals. It is then run at no load for a period of fifteen to thirty minutes, in order to obtain uniform

conditions of friction, etc. If, now, the armature current is read, this furnishes the required value of VA_0.

If the motor is equipped with a rheostatic control, giving it different speeds, the no load value of the armature watts. AV_0, should be determined at each different speed at which the motor is to be run, and a speed-ampere curve drawn. Then, when the motor is operating under load, it is necessary not only to read the armature amperes, but also the speed in order to apply the corresponding value of AV_0 in the B.h.p. formula.

(b) **Efficiency.** If this is required, it is only necessary to obtain the total watts input. This divided into the watts output equals the efficiency. For a shunt wound constant speed motor, the field current, being approximately constant, may be predetermined and added as a constant to the armature current to get the total amperes input.

Problem 74_1. At no load, a motor takes 10.5 amperes armature current at 220 volts. What is the value of AV_0? Is this what is referred to as the "constant" loss? What is the constant loss, in watts (see Problem 74_3)?
Ans., 2310 watts.

Problem 74_2. Same motor, given the armature resistance = 0.038 ohm. What is the brake horse-power, when the armature current is 150 amperes?
Ans., 40 B.h.p.

Problem 74_3. What is the efficiency of the preceding, if the field current is 5.8 amperes. *Ans.*, 87 per cent.

APPENDIX A

LOGARITHMS

To Find the Fractional Power of a Number Less than Unity.

To avoid the use of negative characteristics, the following rule is suggested.

Rule. Express the given number as a fraction whose numerator is unity. Find the required power of the denominator of this fraction, and then reduce to a decimal.

For example,

To find $0.787^{1.33}$,

$$.787=\frac{1}{1.27}$$

$$\text{Log. } 1.27^{1.33}=1.33\times 0.104=0.138;$$

$$0.138=\log 1.37;$$

$$\therefore\quad 1.27^{1.33}=1.37,$$

$$\text{and}\qquad .787^{1.33}=\frac{1}{1.27^{1.33}}=\frac{1}{1.37}$$

$$=0.73.$$

To Find the Napierian Logarithm (Base e = 2.718) of a Number.

Rule. Multiply the common logarithm (base 10) by the constant 2.302, or 2.3, approximately.

For example,

To find $\log_e$ of 315, (see page 366).

$$\text{Log}_{10}\ 315=2.498,$$

$$\therefore\quad \text{Log}_e\ 315=2.3\times 2.498=5.750.$$

COMMON LOGARITHMS

No.	0	1	2	3	4	5	6	7	8	9	Dif.
0	0	0000	3010	4771	6021	6990	7782	8451	9031	9542	
10	0000	0043	0086	0128	0170	0212	0253	0294	0334	0374	42
11	0414	0453	0492	0531	0569	0607	0645	0682	0719	0755	38
12	0792	0828	0864	0899	0934	0969	1004	1038	1072	1106	35
13	1139	1173	1206	1239	1271	1303	1335	1367	1399	1430	32
14	1461	1492	1523	1553	1584	1614	1644	1673	1703	1732	30
15	1761	1790	1818	1847	1875	1903	1931	1959	1987	2014	28
16	2041	2068	2095	2122	2148	2175	2201	2227	2253	2279	26
17	2304	2330	2355	2380	2405	2430	2455	2480	2504	2529	25
18	2553	2577	2601	2625	2648	2672	2695	2718	2742	2765	24
19	2788	2810	2833	2856	2878	2900	2923	2945	2967	2989	22
20	3010	3032	3054	3075	3096	3118	3139	3160	3181	3201	21
21	3222	3243	3263	3284	3304	3324	3345	3365	3385	3404	20
22	3424	3444	3464	3483	3502	3522	3541	3560	3579	3598	19
23	3617	3636	3655	3674	3692	3711	3729	3747	3766	3784	19
24	3802	3820	3838	3856	3874	3892	3909	3927	3945	3962	18
25	3979	3997	4014	4031	4048	4065	4082	4099	4116	4133	17
26	4150	4166	4183	4200	4216	4232	4249	4265	4281	4298	16
27	4314	4330	4346	4362	4378	4393	4409	4425	4440	4456	16
28	4472	4487	4502	4518	4533	4548	4564	4579	4594	4609	15
29	4624	4639	4654	4669	4683	4698	4713	4728	4742	4757	15
30	4771	4786	4800	4814	4829	4843	4857	4871	4886	4900	14
31	4914	4928	4942	4955	4969	4983	4997	5011	5024	5038	14
32	5051	5065	5079	5092	5105	5119	5132	5145	5159	5172	13
33	5185	5198	5211	5224	5237	5250	5263	5276	5289	5302	13
34	5315	5328	5340	5353	5366	5378	5391	5403	5416	5428	13
35	5441	5453	5465	5478	5490	5502	5514	5527	5539	5551	12
36	5563	5575	5587	5599	5611	5623	5635	5647	5658	5670	12
37	5682	5694	5705	5717	5729	5740	5752	5763	5775	5786	12
38	5798	5809	5821	5832	5843	5855	5866	5877	5888	5899	11
39	5911	5922	5933	5944	5955	5966	5977	5988	5999	6010	11
40	6021	6031	6042	6053	6064	6075	6085	6096	6107	6117	11
41	6128	6138	6149	6160	6170	6180	6191	6201	6212	6222	10
42	6232	6243	6253	6263	6274	6284	6294	6304	6314	6325	10
43	6335	6345	6355	6365	6375	6385	6395	6405	6415	6425	10
44	6435	6444	6454	6464	6474	6484	6493	6503	6513	6522	10
45	6532	6542	6551	6561	6571	6580	6590	6599	6609	6618	10
46	6628	6637	6646	6656	6665	6675	6684	6693	6702	6712	9
47	6721	6730	6739	6749	6758	6767	6776	6785	6794	6803	9
48	6812	6821	6830	6839	6848	6857	6866	6875	6884	6893	9
49	6902	6911	6920	6928	6937	6946	6955	6964	6972	6981	9
50	6990	6998	7007	7016	7024	7033	7042	7050	7059	7067	9
51	7076	7084	7093	7101	7110	7118	7126	7135	7143	7152	9
52	7160	7168	7177	7185	7193	7202	7210	7218	7226	7235	8
53	7243	7251	7259	7267	7275	7284	7292	7300	7308	7316	8
54	7324	7332	7340	7348	7356	7364	7372	7380	7388	7396	8

COMMON LOGARITHMS

No.	0	1	2	3	4	5	6	7	8	9	Dif.
55	7404	7412	7419	7427	7435	7443	7451	7459	7466	7474	8
56	7482	7490	7497	7505	7513	7520	7528	7536	7543	7551	8
57	7559	7566	7574	7582	7589	7597	7604	7612	7619	7627	8
58	7634	7642	7649	7657	7664	7672	7679	7686	7694	7701	7
59	7709	7716	7723	7731	7738	7745	7752	7760	7767	7774	7
60	7782	7789	7796	7803	7810	7818	7825	7832	7839	7846	7
61	7853	7860	7868	7875	7882	7889	7896	7903	7910	7917	7
62	7924	7931	7938	7945	7952	7959	7966	7973	7980	7987	7
63	7993	8000	8007	8014	8021	8028	8035	8041	8048	8055	7
64	8062	8069	8075	8082	8089	8096	8102	8109	8116	8122	7
65	8129	8135	8142	8149	8156	8162	8169	8176	8182	8189	7
66	8195	8202	8209	8215	8222	8228	8235	8241	8248	8254	7
67	8261	8267	8274	8280	8287	8293	8299	8306	8312	8319	6
68	8325	8331	8338	8344	8351	8357	8363	8370	8376	8382	6
69	8388	8395	8401	8407	8414	8420	8426	8432	8439	8445	6
70	8451	8457	8463	8470	8476	8482	8488	8494	8500	8506	6
71	8513	8519	8525	8531	8537	8543	8549	8555	8561	8567	6
72	8573	8579	8585	8591	8597	8603	8609	8615	8621	8627	6
73	8633	8639	8645	8651	8657	8663	8669	8675	8681	8686	6
74	8692	8698	8704	8710	8716	8722	8727	8733	8739	8745	6
75	8751	8756	8762	8768	8774	8779	8785	8791	8797	8802	6
76	8808	8814	8820	8825	8831	8837	8842	8848	8854	8859	6
77	8865	8871	8876	8882	8887	8893	8899	8904	8910	8915	6
78	8921	8927	8932	8938	8943	8949	8954	8960	8965	8971	6
79	8976	8982	8987	8993	8998	9004	9009	9015	9020	9025	5
80	9031	9036	9042	9047	9053	9058	9063	9069	9074	9079	5
81	9085	9090	9096	9101	9106	9112	9117	9122	9128	9133	5
82	9138	9143	9149	9154	9159	9165	9170	9175	9180	9186	5
83	9191	9196	9201	9206	9212	9217	9222	9227	9232	9238	5
84	9243	9248	9253	9258	9263	9269	9274	9279	9284	9289	5
85	9294	9299	9304	9309	9315	9320	9325	9330	9335	9340	5
86	9345	9350	9355	9360	9365	9370	9375	9380	9385	9390	5
87	9395	9400	9405	9410	9415	9420	9425	9430	9435	9440	5
88	9445	9450	9455	9460	9465	9469	9474	9479	9484	9489	5
89	9494	9499	9504	9509	9513	9518	9523	9528	9533	9538	5
90	9542	9547	9552	9557	9562	9566	9571	9576	9581	9586	5
91	9590	9595	9600	9605	9609	9614	9619	9624	9628	9633	5
92	9638	9643	9647	9652	9657	9661	9666	9671	9675	9680	5
93	9685	9689	9694	9699	9703	9708	9713	9717	9722	9727	5
94	9731	9736	9741	9745	9750	9754	9759	9763	9768	9773	5
95	9777	9782	9786	9791	9795	9800	9805	9809	9814	9818	5
96	9823	9827	9832	9836	9841	9845	9850	9854	9859	9863	4
97	9868	9872	9877	9881	9886	9890	9894	9899	9903	9908	4
98	9912	9917	9921	9926	9930	9934	9939	9943	9948	9952	4
99	9956	9961	9965	9969	9974	9978	9983	9987	9991	9996	4

DIAMETERS AND AREAS OF CIRCLES

Diam.	Area.	Diam.	Area.	Diam.	Area.	Diam.	Area.	Diam.	Area.
1/16	.00307	15/16	6.78	13/16	26.5	3/8	102.	1/8	230.
1/8	.0123	3.	7.07	7/8	27.1	1/2	104.	1/4	234.
3/16	.0276	1/16	7.37	15/16	27.7	5/8	106.	3/8	237.
1/4	.0491	1/8	7.67	6.	28.3	3/4	108.	1/2	240.
5/16	.0767	3/16	7.98	1/8	29.5	7/8	111.	5/8	244.
3/8	.110	1/4	8.30	1/4	30.7	12.	113.	3/4	247.
7/16	.150	5/16	8.62	3/8	31.9	1/8	115.	7/8	251.
1/2	.196	3/8	8.95	1/2	33.2	1/4	118.	18.	254.
9/16	.248	7/16	9.28	5/8	34.5	3/8	120.	1/8	258.
5/8	.307	1/2	9.62	3/4	35.8	1/2	123.	1/4	262.
11/16	.371	9/16	9.97	7/8	37.1	5/8	125.	3/8	265.
3/4	.442	5/8	10.3	7.	38.5	3/4	128.	1/2	269.
13/16	.518	11/16	10.7	1/8	39.9	7/8	130.	5/8	272.
7/8	.601	3/4	11.0	1/4	41.3	13.	133.	3/4	276.
15/16	.690	13/16	11.4	3/8	42.7	1/8	135.	7/8	280.
1.	.785	7/8	11.8	1/2	44.2	1/4	138.	19.	283.
1/16	.887	15/16	12.2	5/8	45.7	3/8	140.	1/8	287.
1/8	.994	4.	12.6	3/4	47.2	1/2	143.	1/4	291.
3/16	1.11	1/16	13.0	7/8	48.7	5/8	146.	3/8	295.
1/4	1.23	1/8	13.4	8.	50.3	3/4	148.	1/2	299.
5/16	1.35	3/16	13.8	1/8	51.8	7/8	151.	5/8	302.
3/8	1.48	1/4	14.2	1/4	53.5	14.	154.	3/4	306.
7/16	1.62	5/16	14.6	3/8	55.1	1/8	157.	7/8	310.
1/2	1.77	3/8	15.0	1/2	56.7	1/4	159.	20.	314.
9/16	1.92	7/16	15.5	5/8	58.4	3/8	162.	1/8	318.
5/8	2.07	1/2	15.9	3/4	60.1	1/2	165.	1/4	322.
11/16	2.24	9/16	16.3	7/8	61.9	5/8	168.	3/8	326.
3/4	2.40	5/8	16.8	9.	63.6	3/4	171.	1/2	330.
13/16	2.58	11/16	17.3	1/8	65.4	7/8	174.	5/8	334.
7/8	2.76	3/4	17.7	1/4	67.2	15.	177.	3/4	338.
15/16	2.95	13/16	18.2	3/8	69.0	1/8	180.	7/8	342.
2.	3.14	7/8	18.7	1/2	70.9	1/4	183.	21.	346.
1/16	3.34	15/16	19.1	5/8	72.8	3/8	186.	1/8	350.
1/8	3.55	5.	19.6	3/4	74.7	1/2	189.	1/4	355.
3/16	3.76	1/16	20.1	7/8	76.6	5/8	192.	3/8	359.
1/4	3.98	1/8	20.6	10.	78.5	3/4	195.	1/2	363.
5/16	4.20	3/16	21.1	1/8	80.5	7/8	198.	5/8	367.
3/8	4.43	1/4	21.6	1/4	82.5	16.	201.	3/4	371.
7/16	4.67	5/16	22.2	3/8	84.5	1/8	204.	7/8	376.
1/2	4.91	3/8	22.7	1/2	86.6	1/4	207.	22.	380.
9/16	5.16	7/16	23.2	5/8	88.7	3/8	211.	1/8	384.
5/8	5.41	1/2	23.8	3/4	90.8	1/2	214.	1/4	389.
11/16	5.67	9/16	24.3	7/8	92.9	5/8	217.	3/8	393.
3/4	5.94	5/8	24.8	11.	95.0	3/4	220.	1/2	398.
13/16	6.21	11/16	25.4	1/8	97.2	7/8	224.	5/8	402.
7/8	6.49	3/4	26.0	1/4	99.4	17.	227.	3/4	406.

DIAMETERS AND AREAS OF CIRCLES—*Continued*

Diam.	Area.	Diam.	Area.	Diam.	Area	Diam.	Area.	Diam.	Area.
$\frac{7}{8}$	411.	$\frac{3}{8}$	466.	$\frac{7}{8}$	525.	$\frac{3}{8}$	588.	$\frac{7}{8}$	654.
23.	415.	$\frac{1}{2}$	471.	26.	531.	$\frac{1}{2}$	594.	29.	660.
$\frac{1}{8}$	420.	$\frac{5}{8}$	476.	$\frac{1}{8}$	536.	$\frac{5}{8}$	599.	$\frac{1}{8}$	666.
$\frac{1}{4}$	425.	$\frac{3}{4}$	481.	$\frac{1}{4}$	541.	$\frac{3}{4}$	605.	$\frac{1}{4}$	672.
$\frac{3}{8}$	429.	$\frac{7}{8}$	485.	$\frac{3}{8}$	546.	$\frac{7}{8}$	610.	$\frac{3}{8}$	677.
$\frac{1}{2}$	434.	25.	491.	$\frac{1}{2}$	551.	28.	616.	$\frac{1}{2}$	683.
$\frac{5}{8}$	438.	$\frac{1}{8}$	495.	$\frac{5}{8}$	556.	$\frac{1}{8}$	621.	$\frac{5}{8}$	689.
$\frac{3}{4}$	443.	$\frac{1}{4}$	501.	$\frac{3}{4}$	562.	$\frac{1}{4}$	627.	$\frac{3}{4}$	695.
$\frac{7}{8}$	448.	$\frac{3}{8}$	505.	$\frac{7}{8}$	567.	$\frac{3}{8}$	632.	$\frac{7}{8}$	700.
24.	452.	$\frac{1}{2}$	511.	27.	573.	$\frac{1}{2}$	638.	30.	707.
$\frac{1}{8}$	457.	$\frac{5}{8}$	515.	$\frac{1}{8}$	577.	$\frac{5}{8}$	643.		
$\frac{1}{4}$	462.	$\frac{3}{4}$	521.	$\frac{1}{4}$	583	$\frac{3}{4}$	649.		

WEIGHT OF ONE CUBIC FOOT OF WATER AT VARIOUS TEMPERATURES

Temp., Deg. F.	Weight, Lbs. per Cu. Ft.	Temp., Deg. F.	Weight, Lbs. per Cu. Ft.
30	62.42	190	60.36
40	62.43	200	60.12
50	62.42	210	59.88
60	62.37	220	59.63
70	62.30	230	59.37
80	62.22	240	59.11
90	62.11	250	58.83
100	62.00	260	58.55
110	61.86	270	58.26
120	61.71	280	57.96
130	61.55	290	57.65
140	61.38	300	57.33
150	61.20	310	57.00
160	61.00	320	56.66
170	60.80	330	56.30
180	60.58	340	55.94

STEAM TABLES FOR CONDENSER CALCULATIONS

Revised from Marks' Mechanical Engineers' Handbook.

Temperature Deg. Fahr.	Vacuum in In. of Mercury Referred to a 30-in. Bar. (Mercury at 58.4° F.)	Pressure Lb. Per Sq. In. Absolute	Specific Volume Cu. Ft. Per Lb.	Heat of the Liquid	Total Heat of the Steam
50	29.637	0.1780	1702.0	18.08	1081.4
52	29.609	0.1917	1586.0	20.08	1082.3
54	29.579	0.2063	1480.0	22.08	1083.2
56	29.547	0.2219	1381.0	24.08	1084.1
58	29.513	0.2385	1291.0	26.08	1085.0
60	29.477	0.2562	1208.0	28.08	1085.9
62	29.439	0.2749	1130.0	30.08	1086.8
64	29.398	0.2949	1058.0	32.07	1087.6
66	29.354	0.3161	991.0	34.07	1088.5
68	29.308	0.3386	928.0	36.07	1089.4
70	29.259	0.3626	871.0	38.06	1090.3
72	29.208	0.3880	817.0	40.05	1091.2
74	29.153	0.4148	767.0	42.05	1092.1
76	29.095	0.4432	720.0	44.04	1093.0
78	29.034	0.4735	677.0	46.04	1093.9
80	28.968	0.505	636.8	48.03	1094.8
82	28.899	0.539	598.7	50.03	1095.6
84	28.826	0.575	562.9	52.02	1096.5
86	28.749	0.613	529.5	54.01	1097.4
88	28.666	0.654	498.4	56.01	1098.3
90	28.580	0.696	469.3	58.00	1099.2
92	28.489	0.741	442.2	60.00	1100.1
94	28.392	0.789	417.0	61.99	1101.0
96	28.290	0.838	393.4	63.98	1101.8
98	28.183	0.891	371.4	65.98	1102.8
100	28.070	0.946	350.8	67.97	1103.6
102	27.951	1.005	331.5	69.96	1104.5
104	27.825	1.066	313.3	71.96	1105.3
106	27.692	1.131	296.4	73.95	1106.2
108	27.550	1.199	280.5	75.95	1107.1
110	27.404	1.271	265.5	77.94	1108.0
112	27.250	1.346	251.4	79.93	1108.8
114	27.088	1.426	238.2	81.93	1109.7
116	26.919	1.509	225.8	83.92	1110.6
118	26.739	1.597	214.1	85.92	1111.5
120	26.553	1.689	203.1	87.91	1112.3
122	26.355	1.785	192.8	89.91	1113.2
124	26.149	1.886	183.1	91.90	1114.1
126	25.931	1.992	173.9	93.90	1115.0
128	25.706	2.103	165.3	95.89	1115.8
130	25.48	2.219	157.1	97.89	1116.7
135	24.84	2.53	138.7	102.9	1118.8
140	24.12	2.88	122.8	107.9	1121.0
145	23.33	3.28	109.0	112.9	1123.1
150	22.43	3.71	96.9	117.9	1125.3

PROPERTIES OF SATURATED STEAM

REPRODUCED FROM MARKS AND DAVIS' "STEAM TABLES AND DIAGRAMS"

Pressure, Pounds Absolute.	Temperature Deg. F.	Density Lbs. per Cu. Ft.	Specific Volume Cu. Ft. per Pound	Heat of the Liquid B.t.u. h	Latent Heat of Evap. B.t.u. L	Total Heat of Steam B.t.u. H
1	101.83	.00300	333.0	69.8	1034.6	1104.4
2	126.15	.00576	173.5	94.0	1021.0	1115.0
3	141.52	.00845	118.5	109.4	1012.3	1121.6
4	153.01	.0111	90.5	120.9	1005.7	1126.5
5	162.28	.0136	73.33	130.1	1000.3	1130.5
6	170.06	.0162	61.89	137.9	995.8	1133.7
7	176.85	.0187	53.56	144.7	991.8	1136.5
8	182.86	.0211	47.27	150.8	988.2	1139.0
9	188.27	.0236	42.36	156.2	985.0	1141.1
10	193.22	.0261	38.38	161.1	982.0	1143.1
11	197.75	.0285	35.10	165.7	979.2	1144.9
12	201.96	.0309	32.36	169.9	976.6	1146.5
13	205.87	.0333	30.03	173.8	974.2	1148.0
14	209.55	.0357	28.02	177.5	971.9	1149.4
14.7	212	.0375	26.73	180	970.4	1150.4
15	213.0	.0381	26.27	181.0	969.7	1150.7
16	216.3	.0404	24.79	184.4	967.6	1152.0
17	219.4	.0428	23.38	187.5	965.6	1153.1
18	222.4	.0451	22.16	190.5	963.7	1154.2
19	225.2	.0475	21.07	193.4	961.8	1155.2
20	228.0	.0498	20.08	196.1	960.0	1156.2
22	233.1	.0545	18.37	201.3	956.7	1158.0
24	237.8	.0591	16.93	206.1	953.5	1159.6
26	242.2	.0636	15.72	210.6	950.6	1161.2
28	246.4	.0682	14.67	214.8	947.8	1162.6
30	250.3	.0728	13.74	218.8	945.1	1163.9
32	254.1	.0773	12.93	222.6	942.5	1165.1
34	257.6	.0818	12.22	226.2	940.1	1166.3
36	261.0	.0863	11.58	229.6	937.7	1167.3
38	264.2	.0908	11.01	232.9	935.5	1168.4
40	267.3	.0953	10.49	236.1	933.3	1169.4
42	270.2	.0998	10.02	239.1	931.2	1170.3
44	273.1	.104	9.59	242.0	929.2	1171.2
46	275.8	.109	9.20	244.8	927.2	1172.0
48	278.5	.113	8.84	247.5	925.3	1172.8

PROPERTIES OF SATURATED STEAM

Pressure, Pounds Absolute.	Temperature Deg. F.	Density Lbs. per Cu. Ft.	Specific Volume Cu. Ft. Per Pound.	Heat of the Liquid B.t.u. h	Latent Heat of Evap. B.t.u. L	Total Heat of Steam B.t.u. H
50	281.0	.117	8.51	250.1	923.5	1173.6
52	283.5	.122	8.20	252.6	921.7	1174.3
54	285.9	.126	7.91	255.1	919.9	1175.0
56	288.2	.131	7.65	257.5	918.2	1175.7
58	290.5	.135	7.40	259.8	916.5	1176.4
60	292.7	.139	7.17	262.1	914.9	1177.0
62	294.9	.144	6.95	264.3	913.3	1177.6
64	297.0	.148	6.75	266.4	911.8	1178.2
66	299.0	.152	6.56	268.5	910.2	1178.8
68	301.0	.157	6.38	270.6	908.7	1179.3
70	302.9	.161	6.20	272.6	907.2	1179.8
72	304.8	.166	6.04	274.5	905.8	1180.4
74	306.7	.170	5.89	276.5	904.4	1180.9
76	308.5	.174	5.74	278.3	903.0	1181.4
78	310.3	.179	5.60	280.2	901.7	1181.8
80	312.0	.183	5.47	282.0	900.3	1182.3
82	313.8	.187	5.34	283.8	899.0	1182.8
84	315.4	.191	5.22	285.5	897.7	1183.2
86	317.1	.196	5.10	287.2	896.4	1183.6
88	318.7	.200	5.00	288.9	895.2	1184.0
90	320.3	.204	4.89	290.5	893.9	1184.4
92	321.8	.209	4.79	292.1	892.7	1184.8
94	323.4	.213	4.69	293.7	891.5	1185.2
96	324.9	.217	4.60	295.3	890.3	1185.6
98	326.4	.221	4.51	296.8	889.2	1186.0
100	327.8	.226	4.429	298.3	888.0	1186.3
105	331.4	.236	4.230	302.0	885.2	1187.2
110	334.8	.247	4.047	305.5	882.5	1188.0
115	338.1	.258	3.880	309.0	879.8	1188.8
120	341.3	.268	3.726	312.3	877.2	1189.6
125	344.4	.279	3.583	315.5	874.7	1190.3
130	347.4	.290	3.452	318.6	872.3	1191.0
135	350.3	.300	3.331	321.7	869.9	1191.6
140	353.1	.311	3.219	324.6	867.6	1192.2
145	355.8	.321	3.112	327.4	865.4	1192.8

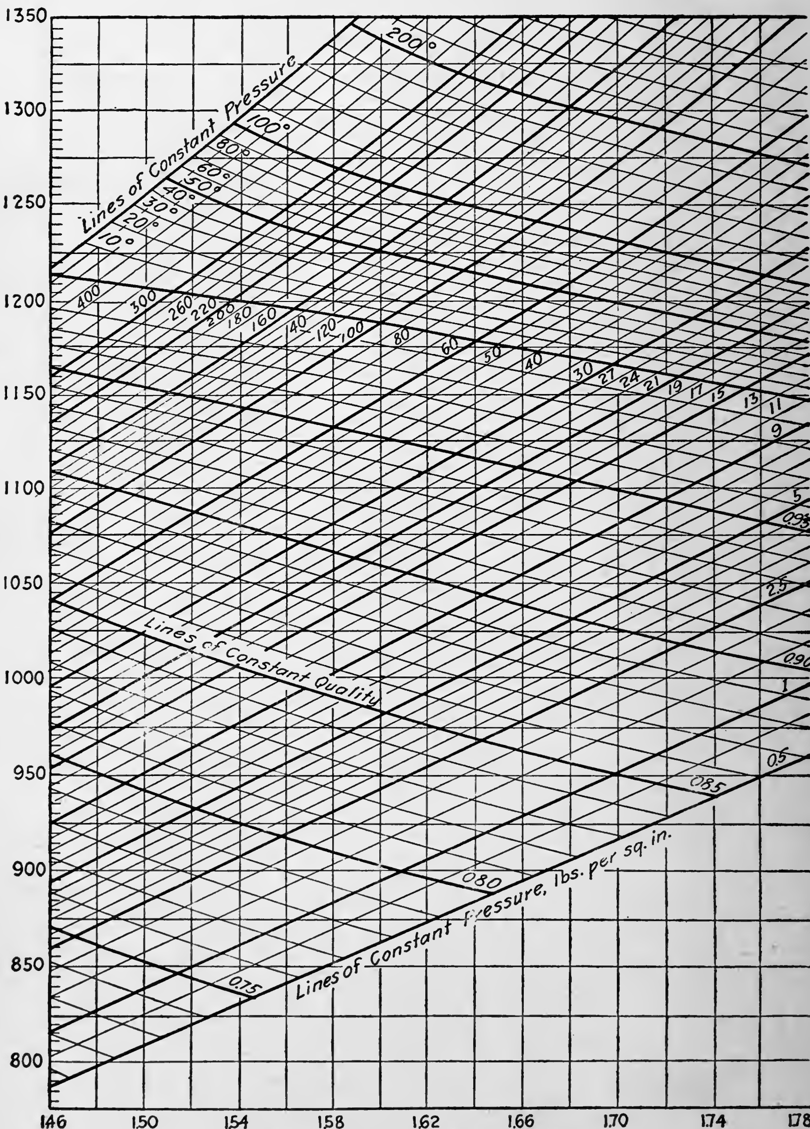
PROPERTIES OF SATURATED STEAM

Pressure Pounds Absolute.	Temperature Deg. F.	Density Lbs. per Cu. Ft.	Specific Volume Cu. Ft. per Pound.	Heat of the Liquid, B.t.u. *h*	Latent Heat of Evap., B.t.u. *L*	Total Heat of Steam, B.t.u. *H*
150	358.5	.332	3.012	330.2	863.2	1193.4
155	361.0	.342	2.920	332.9	861.0	1194.0
160	363.6	.353	2.834	335.6	858.8	1194.5
165	366.0	.363	2.753	338.2	856.8	1195.0
170	368.5	.374	2.675	340.7	854.7	1195.4
175	370.8	.384	2.602	343.2	852.7	1195.9
180	373.1	.395	2.553	345.6	850.8	1196.4
185	375.4	.405	2.468	348.0	848.8	1196.8
190	377.6	.416	2.406	350.4	846.9	1197.3
195	379.8	.426	2.346	352.7	845.0	1197.7
200	381.9	.437	2.290	354.9	843.2	1198.1
205	384.0	.447	2.237	357.1	841.4	1198.5
210	386.0	.457	2.187	359.2	839.6	1198.8
215	388.0	.468	2.138	361.4	837.9	1199.2
220	389.9	.478	2.091	363.4	836.2	1199.6
225	391.9	.489	2.046	365.5	834.4	1199.9
230	393.8	.499	2.004	367.5	832.8	1200.2
235	395.6	.509	1.964	369.4	831.1	1200.6
240	397.4	.520	1.924	371.4	829.5	1200.9
245	399.3	.530	1.887	373.3	827.9	1201.2
250	401.1	.541	1.850	375.2	826.3	1201.5

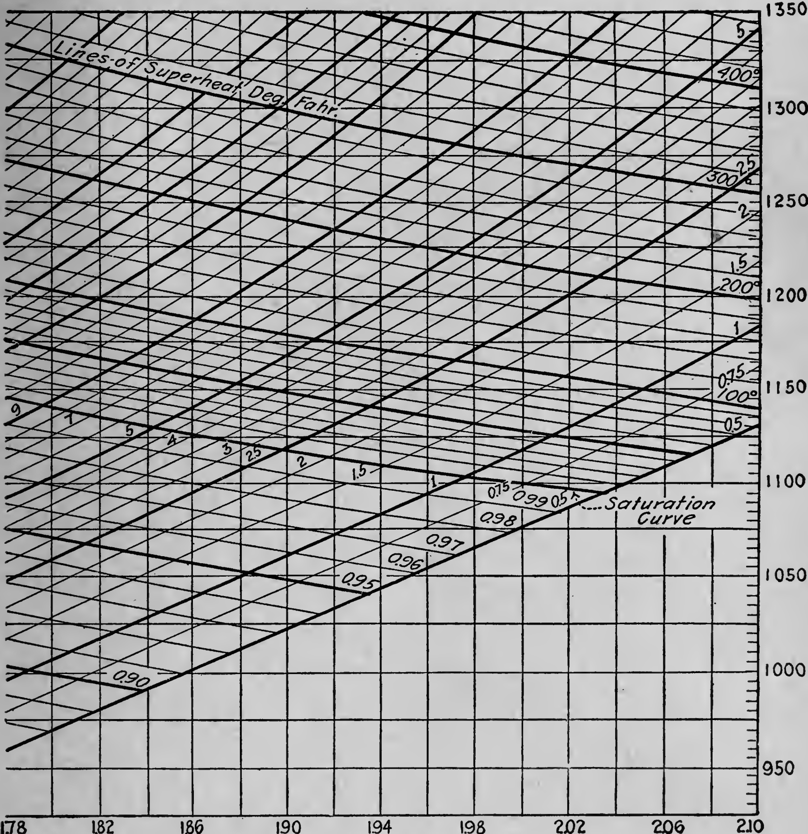
MEAN SPECIFIC HEAT OF SUPERHEATED STEAM

CALCULATED FROM MARKS AND DAVIS TABLES

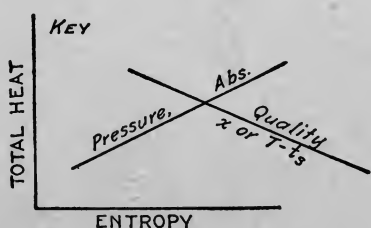
Gauge Pressure	Degree of Superheat															
	50	60	70	80	90	100	110	120	130	140	150	160	170	180	190	200
50	.518	.517	.514	.513	.511	.510	.508	.507	.505	.504	.503	.502	.501	.500	.500	.499
60	.528	.525	.523	.521	.519	.517	.515	.513	.512	.511	.509	.508	.507	.506	.504	.504
70	.536	.534	.531	.529	.527	.524	.522	.520	.518	.516	.515	.513	.512	.511	.510	.509
80	.544	.542	.539	.535	.532	.530	.528	.526	.524	.522	.520	.518	.516	.515	.514	.513
90	.553	.550	.546	.543	.539	.536	.534	.532	.529	.527	.525	.523	.521	.519	.518	.517
100	.562	.557	.553	.549	.544	.542	.539	.536	.533	.531	.529	.527	.525	.523	.522	.521
110	.570	.565	.560	.556	.552	.548	.545	.542	.539	.536	.534	.532	.529	.528	.526	.525
120	.578	.573	.567	.561	.557	.554	.550	.546	.543	.540	.537	.535	.533	.531	.529	.528
130	.586	.580	.574	.569	.564	.560	.555	.552	.548	.545	.542	.539	.537	.535	.533	.531
140	.594	.588	.581	.575	.570	.565	.561	.557	.553	.550	.547	.544	.541	.539	.536	.534
150	.604	.595	.587	.581	.576	.570	.566	.561	.557	.554	.550	.547	.544	.542	.539	.537
160	.612	.603	.596	.589	.582	.576	.571	.566	.562	.558	.554	.551	.548	.545	.543	.541



MOLLIER DIAGRAM (STEAM)



MOLLIER DIAGRAM (STEAM)



See page 378 for explanation.

MOLLIER DIAGRAM—EXPLANATION

Total heat of steam, H, is plotted against corresponding values of entropy. The condition of the steam as regards pressure and quality are denoted by diagonal lines (see key, page 377). Any two of these diagonals intersect at a point which establishes the total heat and entropy corresponding to the condition.

Example 1. Find the total heat of steam at 100 lbs. pressure absolute, and a quality of 95 per cent.

First, determine the intersection of the 100 lb. pressure line with the .95 quality line. At the intersection, read the ordinate value, 1142 B.t.u.

Example 2. Find the total heat of steam at 100 lbs. pressure, absolute, and a temperature of 378°.

From the steam table, p. 372, it is found that the given temperature is 50° higher than the temperature of saturation at 100 lbs. Referring to the Mollier Diagram, find the intersection of the 100 lb. absolute line with the 50° superheat line. At this point read the ordinate value, 1215 B.t.u.

Example 3. Find the condition of steam after it has expanded adiabatically from 100 lbs. absolute and 50° superheat to a final pressure of 15 lbs. absolute.

The intersection of the 100 lb. line and the 50° superheat line, on the chart, corresponds to an entropy of 1.636. Find where this entropy line intersects the 15 lb. curve. The corresponding quality is seen to be .916 and the total heat, 1070 B.t.u.

Note that the difference between the initial and final heats, namely $1215-1070=145$ B.t.u., is the heat available to do work on the Rankine cycle. This method is therefore useful to calculate the Rankine efficiency (see p. 235).

Example 4. Find the quality of steam from a throttling calorimeter determination, data as on page 146.

Take the calorimeter pressure as 15 lbs. and the absolute (high) steam pressure as 115 lbs. The superheat in the calorimeter

chamber is 10°. Find the intersection of the 15 lb. line, on the chart, with the 10° superheat line. The corresponding total heat is 1157 B.t.u. Find where the 1157 total heat line intersects the 115 lb. line (tracing horizontally to the left). This intersection represents the condition of the steam before expanding, and the quality, from the chart, is .961.

NOTE. When scaling the diagram and tracing constant entropy or constant total heat lines, a pair of dividers will be found convenient.

EXAMPLE 5. To find the quality x of the exhaust steam from an engine. Let H be the total heat of the steam entering, and H_e that of the exhaust. All of the heat, H, appears in the exhaust except the amounts turned into work, and radiated from the engine cylinder. For a steam turbine, radiation may be taken between $.001H$ and $.005H$; for a reciprocating engine between $.003H$ and $.02H$. Call this coefficient of H, k. Then, referring to page 229 for "heat converted into work,"

$$H_e=(1-k)H-2545/S.$$

Having found H_e, locate its intersection of the total heat line, H_e, with the exhaust pressure line. At this intersection read the quality.

Taking, for example, $H=1192$, $k=.01$, $S=31.8$, and atmospheric exhaust

$$H_e=.98\times 1192-2545/31.8=1100 \text{ B.t.u.}$$

The 1100 B.t.u. line on the chart intersects the 15 lb. line at a quality of 94.7 per cent.

PROPERTIES OF AMMONIA

(Goodenough and Mosher)

Temp. Deg. Fahr.	Pressure, Lb. per Sq. In., Abs.	Specific Volume		Heat Content		Heat of Vaporization.
		of Liquid Cu. Ft. per Lb.	of Sat. Vapor Cu. Ft. per Lb.	of Liquid.	of Sat. Vapor.	
−40	10.12	0.0234	25.45	−75.3	526.6	601.9
−35	11.74	0.0235	22.14	−70.2	528.2	598.3
−30	13.56	0.0236	19.35	−65.0	529.8	594.7
−25	15.61	0.0238	16.95	−59.8	531.3	591.1
−20	17.91	0.0239	14.89	−54.6	532.8	587.4
−15	20.46	0.0240	13.15	−49.4	534.3	583.6
−10	23.30	0.0241	11.63	−44.2	535.7	579.9
− 5	26.46	0.0242	10.32	−38.9	537.1	576.1
0	29.95	0.0244	9.19	−33.7	538.5	572.2
5	33.79	0.0245	8.20	−28.4	539.9	568.3
10	38.02	0.0246	7.34	−23.2	541.2	564.4
15	42.67	0.0248	6.583	−17.9	542.5	560.4
20	47.75	0.0249	5.920	−12.6	543.7	556.3
25	53.30	0.0250	5.336	− 7.3	545.0	552.2
30	59.39	0.0252	4.820	− 1.9	546.2	548.1
35	65.91	0.0253	4.364	+ 3.5	547.4	543.9
40	73.03	0.0255	3.959	8.9	548.5	539.7
45	80.75	0.0256	3.599	14.3	549.7	535.3
50	89.09	0.0258	3.278	19.8	550.8	531.0
55	98.03	0.0259	2.992	25.3	551.9	526.5
60	107.7	0.0261	2.734	30.9	552.9	522.0
65	118.1	0.0263	2.503	36.5	554.0	517.5
70	129.2	0.0264	2.296	42.1	555.0	512.8
75	141.1	0.0266	2.109	47.8	556.0	508.1
80	153.9	0.0268	1.940	53.6	557.0	503.4
85	167.4	0.0270	1.788	59.4	557.9	498.5
90	181.8	0.0271	1.650	65.3	558.9	493.5
95	197.3	0.0273	1.524	71.3	559.8	488.5
100	213.8	0.0275	1.408	77.3	560.7	483.4
105	231.2	0.0277	1.305	83.4	561.6	478.2
110	249.6	0.0280	1.210	89.6	562.5	472.9
115	269.2	0.0282	1.122	95.9	563.3	467.4
120	289.9	0.0284	1.042	102.2	564.2	461.9
125	311.6	0.0286	0.970	108.7	565.0	456.3

HYGROMETRY

Hygrometry deals with the determination of the properties of mixtures of water vapor and air. Dalton's Law (p. 142) bears directly upon this subject and should be understood.

It has been shown that a cubic foot of space at a given temperature, *t*, can contain no more than a fixed amount of H_2O vapor, regardless of the presence or absence of any other gas. This maximum amount is the weight of a cubic foot of saturated steam at the existing temperature, *t*.

Supposing that the cubic foot of space contains air at the same time that it contains the maximum amount of H_2O corresponding to the temperature, *t*. Then the air is said to be "saturated," and to have 100 per cent **relative humidity.** If, however, there is less vapor than this present,

$$\text{Per cent, Relative Humidity} = \frac{\text{Weight of water present, pounds per cubic foot}}{\text{Density of saturated steam at } t \text{ degrees}} \times 100.$$

The relative humidity is determined by the "**wet-and-dry bulb**" thermometer, or "psychrometer." This instrument consists of two thermometers, the bulb of one of which is kept wet by surrounding it with a wick saturated with water at room temperature. The evaporation from this wick lowers the temperature of the wet bulb. The dryer is the room air, the lower is the wet bulb temperature; hence the difference between the indications of the two thermometers is a measure of the humidity. The following table, taken from Kent's Mechanical Engineers' Pocketbook, may be used:

RELATIVE HUMIDITY, PER CENT

Dry Thermometer, Deg. F.	Difference between the Dry and Wet Thermometers, Deg. F.																										
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	26	28	30
	Relative Humidity, Saturation being 100. (Barometer = 30 ins.)																										
32	89	79	69	59	49	39	30	20	11	2																	
40	92	83	75	68	60	52	45	37	29	23	15	7	0														
50	93	87	80	74	67	61	55	49	43	38	32	27	21	16	11	5	0										
60	94	89	83	78	73	68	63	58	53	48	43	39	34	30	26	21	17	13	9	5	1						
70	95	90	86	81	77	72	68	64	59	55	51	48	44	40	36	33	29	25	22	19	15	12	9	6			
80	96	91	87	83	79	75	72	68	64	61	57	54	50	47	44	41	38	35	32	29	26	23	20	18	12	7	
90	96	92	89	85	81	78	74	71	68	65	61	58	55	52	49	47	44	41	39	36	34	31	29	26	22	17	13
100	96	93	89	86	83	80	77	73	70	68	65	62	59	56	54	51	49	46	44	41	39	37	35	33	28	24	21
110	97	93	90	87	84	81	78	75	73	70	67	65	62	60	57	55	52	50	48	46	44	42	40	38	34	30	26
120	97	94	91	88	85	82	80	77	74	72	69	67	65	62	60	58	55	53	51	49	47	45	43	41	38	34	31
140	97	95	92	89	87	84	82	79	77	75	73	70	68	66	64	62	60	58	56	54	53	51	49	47	44	41	38

For example, if the wet bulb indicates 60° and the dry bulb 70°, then the difference is 10° and the humidity is 55 per cent.

Calculation of pressures and weights of air, water vapor, and mixture. The following notation will be used, all pressures in pounds per square inch absolute, and weights in pounds per cubic foot.

P_m, W_m = pressure and weight of mixture;
P_a, W_a = partial pressure, and weight of the air;
P_v, W_v = partial pressure and weight of the vapor;
P', W' = pressure and weight of saturated steam at temperature, t;
t = temperature, degrees F., of the mixture;
H = relative humidity, per cent.

To find partial pressures, given total pressure, temperature and humidity. Find P' from the steam tables, corresponding with t.

Then
$$P_v = \frac{H}{100} \times P',$$

and
$$P_a = P_m - P_v.$$

The pressure of the mixture is to be found from the barometer, and the humidity as previously described.

For example, if the barometer reads 29.33″, the corresponding $P_m = 14.41$ lbs. per square inch. Suppose the relative humidity is 55 per cent and the temperature of the mixture is 70°. From the steam tables (p. 370) saturated steam at 70° has a pressure of 0.363 lbs. $(=P')$. Hence, the partial pressure of the vapor in the mixture is

$$P_v = .55 \times 0.363 = 0.2 \text{ lb.},$$

and

$$P_a = 14.41 - .2 = 14.21 \text{ lbs.}$$

To find weights, given partial pressures, temperature and humidity. Find W' (density) from steam tables corresponding to t.

Then

$$W_v = \frac{H}{100} \times W';$$

$$W_a = \frac{144 P_a}{53.4 \times (t+460)}; \quad \text{from } PV = RT;$$

$$W_m = W_v + W_a.$$

For example, using the previous data, the density of saturated steam at 70° is $0.00115 (=W')$.

$$W_v = .55 \times .00115 = .00063;$$

$$W_a = \frac{144 \times 14.21}{53.4 \times 530} = .0725;$$

$$W_m = .00063 + .0725 = .0731.$$

TOTAL HEAT OF AIR–STEAM MIXTURES

Using the notation of the preceding section, it was seen that W' or W_v and W_a, the weights of water vapor and of air per cubic foot of humid air, can be calculated. The heat content or "total heat" per pound of the low pressure steam in the mixture is

$$h'+L', \text{ if 100 per cent humid,}$$
$$\text{or } h_v+L_v+C_p(t-t_s), \text{ if less than 100 per cent humid;}$$

in which $h'+L'$, in the first case, is the total heat at a pressure corresponding to the temperature, t, of the mixture; and, in the second case, h_v+L_v corresponds to the partial pressure of the vapor, P_v, as determined in the preceding section, and t_s is the temperature of saturation of that pressure.

The general condition is that the humidity is less than 100 per cent. As the correct value of total heat given above involves the use of the steam tables, etc., it is preferable to use the closely approximate formula for total heat of superheated steam at low pressures, quoted on page 143, namely

$$H=1058+.455\,t.$$

In this connection t is the temperature, deg. F., of the steam and equal to that of the mixture.

We then have for the total heat of the steam in 1 cu. ft. of the mixture:

$$W'\,(h'+L') \text{ if 100 per cent humid}$$
$$W_v\,(1058+.455\,t) \text{ if less than 100 per cent humid.}$$

The "total heat" of steam is the heat added to H_2O at constant pressure from 32 deg. to bring it to a given condition. Similarly, the "total heat of air" is the heat added at constant

pressure to bring it from 32° to a given condition, t. That is, the total heat of air is

$$C_p \times \text{weight of air} \times (t-32).$$

Taking $C_p = .241$, the total heat of an air-steam mixture in B.t.u. per cubic foot of the mixture is

$$W'(h'+L') + .241\, W_a (t-32) \text{ for 100 per cent humidity}$$
$$W_v (1058 + .455\, t) + .241\, W_a (t-32) \text{ for less than 100 per cent humidity.}$$

Total Heat Values in Gas Combustion. Under "Combustion of Gases" it was shown that, when complete combustion takes place, there is a contraction of volume. Thus, in example worked on page 188, 100 volumes of fuel combined with 521 of air formed 80.2 volumes of CO_2, and 420.8 of N_2 (assuming all of the H_2O to be condensed upon return to standard temperature). In other words, 100 volumes of fuel + 521 of air form 501 of dry products, and the shrinkage is $501 \div 621$. This ratio is called the "coefficient of contraction."

Assume a gas requiring 4.25 cu. ft. of air to burn 1 cu. ft. of the gas. Assume, also, that 25 per cent excess air is used, and that the coefficient of contraction is 0.90. The volumes entering the combustion reaction for 1 cu. ft. of fuel are then

Fuel.................... 1 cu. ft.
Air..................... $4.25 + .25 \times 4.25 = 5.31$ cu. ft.
Products of combustion including excess air....... $5.25 \times .90 + 1.06 = 5.78$ cu. ft.

Application to Junkers Calorimeter Determinations.

Since, in the use of this instrument, the products of combustion, and entering air, and fuel are all at approximately the same temperature, the total heat of the perfect gases entering and leaving will undergo no change. But the air entering combustion may

have any humidity, while the fuel and the products are always 100 per cent humid. Since the products contract in volume there will be a smaller volume carrying away humidity than that carrying humidity in. These two items make necessary a correction, for strict accuracy, thus:

Correction = Total heat of steam in	V_p	cu. ft.	of	products,
– " " "	V_f	"	"	fuel,
– " " "	V_a	"	"	air,

which correction may be plus or minus.

This may be worked conveniently as shown below. The volumes cited above are used as an illustration and it is further assumed that temperatures of room, fuel and products are all 75°, humidity of air entering = 30 per cent and of fuel and products = 100 per cent. Then for the fuel and the products (see steam tables, page 370).

Pressure of the H_2O (at 75°) = .429 lbs. per sq. in.
Weight per cu. ft. of the H_2O (at 75°) = .00135 lbs. per cu. ft.
Total heat per lb. of the H_2O = 1092.5

and for the air

Pressure of the $H_2O = .30 \times .429 = .128$ lbs. per sq. in.
Weight per cu. ft. $= .30 \times .00135 = .000405$ lbs. per cu. ft.
Total heat per lb. $= 1058 + .455 \times 75 = 1092.$

Tabulating the values:

	FUEL	AIR	PRODUCTS
(a) Volumes in cu. ft.	1	5.31	5.78
(b) Weight per cu. ft. of H_2O. . . .	.00135	.000405	.00135
(c) Total heat per lb.	1092.5	1092.0	1092.5
(d) Total heat of total volumes in B.t.u. = (a)×(b)×(c) =	1.47	2.34	8.52

$$\text{Correction} = 8.52 - 1.47 - 2.34 = 4.71 \text{ B.t.u.}$$

REPORTS OF ENGINEERING TESTS

A complete report should describe concisely the experimental object and how it was accomplished, and it should give numerical results and the conclusions formed from them. Whether the report is upon work performed by a student in the laboratory, or by an engineer in practice, its material may be arranged to advantage under the following heads.

(1) Object of the test.
(2) Description and principles of apparatus tested.
(3) Method of testing.
(4) Sample calculations.
(5) Results and curves.
(6) Discussion.
(7) Rough notes or condensed observations.

The subjects under these sub-heads may be treated as follows.

(1) Object of the Experiment. This should be a clear, complete, and concise statement, preferably in one sentence. Its purpose, in practice, is to enable the reader to decide without a full reading whether or not the contents of the report come within his scope of interest. In student work, it should be included as a matter of training. In any case, the object of the test should be stated in writing before its performance in order that the experimenter and others concerned shall have a clear view of the undertaking.

(2) Description and Principles. This should deal only with the machine, instrument, material, or apparatus *tested.* The extent of the treatment is governed by the requirements of those for whom the report is intended. A report prepared as a technical article or address kills itself if it talks over the heads

of its audience or bores them with details with which they are familiar, however much satisfaction it may give the author. This is a matter for judgment. A good rule to follow is to give complete descriptions only of new or little known apparatus; for others it is sufficient to give only commercial sizes and names. It should be remembered, however, that any distinctive feature or any characteristic affecting the test results should be fully described. This depends upon the object of the test. For example, in the report of a mechanical efficiency test of a steam engine, it is appropriate to describe thoroughly anything affecting friction in operation such as lubrication details, balancing of valves, etc. But if the same engine were tested for its steam distribution, lubrication need not be mentioned at all; the valve mechanism then being the important item.

In this division there may be deduced any formulas used in getting results provided they are original or unusual. Otherwise, they may be quoted without deduction, with reference to the authority.

(3) **Method of Testing.** A logical way to begin this subject is by reference to the formulas for the quantities sought. Each formula, reduced to the desired form, shows the quantities that must be directly measured. The means of measuring them may then be described. Usual instruments may be merely mentioned by name, but special apparatus (original, or applicable only to the particular test) should be described fully. Any limitations of apparatus or unusual facilities affecting the precision of measurements should be mentioned to enable the reader to judge for himself the accuracy of the results. For the same reason, the duration of the test should be stated with such items as frequency of readings. In this connection it is a good plan to refer to a sample set of observations which may be included in division 7.

(4) **Sample Calculations.** In practice, these should sometimes be included to make clear the method of testing and as a

voucher of the accuracy of the calculations. They should be omitted if these points do not need amplifying. They should be brief, proceeding at once from the expression for the desired quantity, in which is substituted the test data, to the numerical result. The form under Test 56 (*d*) may be used as a model.

Reports by students should always contain sample calculations. The student should bear in mind that *sample* means something *representative of the whole.* Therefore, if five different quantities are tested for, three times for each one in the same way, five samples are necessary and sufficient. If these tests were repeated by another method involving a different calculation, ten samples are needed. If only one determination is made of each quantity, the sample must be all of the calculations to be representative.

(5) Results and Curves. Numerical results should always be presented in the form of curves when possible, and also as a table. This enables the busy reader to size up the report without fully reading it. Curves and tables should have definite titles and enough information to explain their meaning without reference to the text. Curve sheets should bear the scales of coordinates, and the axes should be scaled for ready reading. Where several curves appear on one sheet, their plotted points should be differentiated by such conventions as circles in outline, solid, half solid, etc. The points should be clearly marked so that they will not be obliterated when the curve is drawn in.

The student should take care to differentiate between *results* and *observations.* Sometimes observations are results, but this is seldom the case. Further, there may be many intermediary quantities between the two which should not be confused with results. If the object of the experiment is clearly stated, it will always show what should comprise the results; in case of doubt, it should be referred to.

(6) Discussion. This should deal with the probable accuracy of the results as affected by the precision of the instruments and

methods used and as indicated by the concordance of the results; it should compare the results with corresponding ones from similar tests, records of which are available in hand-books or elsewhere; and it should give the conclusions to be formed relative to the performance of the apparatus tested and to the physical laws controlling the performance. The conclusions to be formed from a test are the most important part of experimentation, in fact, its very *raison d'être*; so they should be fully given in the report. In practice, there would be no value whatever to a test from which conclusions could not be, or were not, made.

(7) Rough Notes, Observations. Complete reports should contain tables giving the observations in full so that their concordance and validity may be checked by the scientific investigator. When this is not considered necessary, a sample set may be submitted, as for division 3. Students should include in reports the original notes taken in the laboratory, a loose-leaf note-book being used for convenience.

Rough notes should be taken neatly and contain enough data not only to remind the experimenter of the noted quantities, but to enable an outsider without additional explanation to interpret them fully. Inexperienced observers, to save time, are prone to use arbitrary symbols or abbreviations in their notes, without meaning to anyone but themselves. The objections to this practice are that the notes cannot be checked by others and the observer himself is apt to forget their meaning.

In concluding the general subject of reports, it may be well to mention the custom of including in voluminous ones, as a sort of addendum, a synopsis of the whole work, relating briefly the results and conclusions. This enables the busy reader to get the gist of it without a full reading. In technical papers addressed to scientific societies, or published as pamphlets, such a synopsis should conclude the report. If published as an article in a technical journal, it should be at the commencement.

A METHOD FOR CONDUCTING STUDENT TESTS*

The following method has been used with success at Syracuse University.

The first laboratory work assigned is, as far as the equipment will allow, individual. After a little training in the methods of the work, the experiments are given as problems, the theory of which is taught in the classroom. In the laboratory the student is quizzed at intervals during his work, to insure that it be performed not as a matter of rote. Approximate calculations from observations are exacted as the observations are obtained. It is a great mistake to allow the postponement of such calculations until after the test is completed. This point cannot be too strongly emphasized. It is far better to have an experiment only partly but intelligently done than to have a vast amount of observations leading to faulty results and a half-grasp of the principles.

As the course advances, the tests, especially those upon large units, require more than one student to make all the necessary operations and measurements. For the successful conduct of some, it is expedient to have four or five stations at each of which one observer is needed. These advanced tests are undertaken in this way. In the classroom the students solve problems upon the principles involved and are given the special instructions which they could not reasonably be expected to obtain for themselves. When they have shown a satisfactory understanding of the principles, they are allowed to proceed with the experimental work, conducted as follows: For a test requiring five

*Abstracted from a paper by the present writer on "The Teaching of Experimental Engineering," *Educational Review*, June, 1912.

stations, for instance, a squad of six men is selected. For the first fifteen minutes or so of the test, observers A and B are at station No. 1, C at No. 2, D at No. 3, E at No. 4, and F at No. 5. Then B moves to station No. 2, where he is instructed by C in its duties. As soon as B has become familiar with them, C moves on to station No. 3 and is instructed by D concerning the work there, after which D moves to station No. 4 with E, and so on. The scheme can best be understood by the following schedule.

STATION NO.

Time	1	2	3	4	5
10 : 00– : 15	AB	C	D	E	F
: 15– : 20	A	BC	D	E	F
: 20– : 25	A	B	CD	E	F
: 25– : 30	A	B	C	DE	F
: 30– : 35	A	B	C	D	EF
: 35– : 40	FA	B	C	D	E
: 40– : 45	F	AB	C	D	E

Etc.

The advantages of this method over shifting all the students at one time are obvious. There is absolutely no break in the accuracy of the observations or continuity of the test since only one man shifts at a time and he does not make measurements to be used before he has become somewhat familiar with the apparatus. The instructor is relieved of the small but essential details of instruction by the students themselves. Room is made for an additional man on the test without sacrificing the work of any of the others. A complete shift of every man can be made in a shorter time; in the example cited, the interval is thirty minutes.

The time necessary for an observer at a station to instruct the newcomer varies, of course, with the duties of the station. In a boiler test the average time is about ten minutes, so that a complete shift of six men would be effected in sixty minutes.

If all the men shifted at once, the interval should not be less than ninety minutes, preferably two hours.

The shifting is automatic. All that each student needs to know is the sequence of stations, and to remember that he is to move to the next one only when he has been relieved by, and has instructed his successor.

It has been found of considerable advantage to include a station the duty at which is to maintain a "General Log." This contains the observations from all the stations. The student in charge of it checks to some extent these observations upon recording them, and is required also to calculate roughly indicative results as the test proceeds. The log may be used for reference by all the students and enables a clearer view of the whole test.

At the end of a complete shift, three of the men are replaced by three new ones, and put upon the final calculations under the instructor's supervision. When these are completed the resulting quantities are plotted on a large chart, in common use for all who have made the test. In the meantime the test is continued by the three new and the three old observers, the arbitrarily varied quantity of the test having been changed. At the end of the next complete shift, half the men are again replaced, the ones remaining being the more recent ones. In this way each group of three men serves two complete shifts and makes the calculations from the observations of one. When all the results are in, their concordance is checked by the regularity of the plotted curves, and these are presented as a whole to the class for consideration.

APPENDIX B

CODE ON DEFINITIONS AND VALUES

It is by courtesy of the American Society of Mechanical Engineers that this code is printed here. The work of revising the various Power Test Codes is a vast one, and probably will not be completed within two years. The Code on Definitions and Values is the most comprehensive one, but, since it is subject to the decisions of the other Code Committees and the approval of the Society as a whole, its items are still open to some change, as indicated in the comments attached. It is thought, however, that since the recommendations of the Definitions and Values Code Committee are so rational and progressive, they will, with few exceptions, meet with final acceptance. The Code is here printed practically as published, only a few items, not within the scope of this book, being omitted. The arrangement of the tables is slightly different in order to save space.

Credit must be given to the members of the committee, whose names are attached at the end of this reprinting, for their excellent work and fine results.

The Code

The units to be employed in reporting the results of tests made in accordance with the various Power Test Codes are enumerated in Tables 1, 2 and 3. Explanatory and other notes follow in Pars. 101 to 158, to which references are given in the tables.

TABLE 1. FUNDAMENTAL UNITS AND CONSTANTS

NOTE. (ab.) signifies abbreviation; (def.) definition; (a.v.) approximate values.

1 **One foot;** (ab.) ft.; (def.) 12/39.37 of the length of the international prototype meter. (Par. 103.)

2 **One pound mass;** (ab.) lb.; (def.) 0.4535924 times the mass of the international prototype kilogram. (Par. 103.)

3 **Standard gravity;** (ab.) g; (def.) 32.1740 ft./sec.^{2} (Par. 105.)

4 **One pound force**; (ab.) lb.; (def.) a force represented by the weight of one pound mass at a place where gravity has the standard value. (Par. 105.)

5 **One foot-pound**; (ab.) ft-lb.; (def.) the work done by 1 lb. force when its point of application moves one foot in the direction of the force.

6 **One British thermal unit**; (ab.) B.t.u.; (def.) 1/180 of the heat required to raise 1 lb. mass of water from the ice point to the steam point. (Par. 109.)

7 **Absolute temperature**; (ab.) T; (def.) deg. fahr. +459.6 (Par. 108); (a.v.) deg. fahr. +460.

8 **Mechanical equivalent of heat**; (ab.) J; 778 ft-lb. per B.t.u. (equiv.) (Par. 106); (a.v.) 778.

Heat equivalent of work; (ab.) A; 0.001285 B.t.u. per ft-lb. (equiv.) (Par. 106); (a.v.) 0.001285.

9 **One horsepower**; (ab.) hp.; (def.) 550 ft-lb. per sec.; (a.v.) 550. 33,000 ft-lb. per min. (def.); (a.v.) 33,000. 1,980,000 ft-lb. per hr. (def.); (a.v.) 1,980,000. 2,545 B.t.u. per hr. (equiv); (a.v.) 2,545. 745.702 watts (equiv.); (a.v.) 746. 0.7457 kw. (equiv.) (Par. 107); (a.v.) 0.746.

10 **One horsepower-hour**; (ab.) hp-hr.; 2,545 B.t.u. (equiv.) (Par. 107); (a.v.) 2,545.

11 **One kilowatt**; (ab.) kw.; (def.) 1,000 watts; (a.v.) 1,000. 1.3410 hp. (equiv.); (a.v.) 1.341. 3,413 B.t.u. per hr. (equiv.); (a.v.) 3,413. 737.56 ft-lb. per sec. (equiv.) (Par. 107); (a.v.) 387.

12 **One kilowatt-hour**; (ab.) kw-hr.; 1.3410 hp-hr. (equiv.); (a.v.) 1,341. 3,413 B.t.u. (equiv.) (Par. 107); (a.v.) 3,413.

13 **One U. S. gallon**; (ab.) gal.; (def.) 231 cu. in.; (a.v.) 231.

14 **One Standard atmosphere** (International Standard); (ab.) atmos.; (def.) 760 mm. mercury at ice point and standard gravity. (Pars. 108, 109 and 112); (a.v.) 760. 29.9212 in. mercury at ice point and standard gravity (equiv.); (a.v.) 29.92. 14.6963 lb. per sq. in. (equiv.). (Par. 112); (a.v.) 14.7.

15 **One Standard ton refrigeration**; (ab.) ton refr.; (def.) 288,000 B.t.u. (Par. 129); (a.v.) 288,000.

Definitions and values which are of interest to special codes only, are to be found in the "Notes on Data" (Pars. 101–158).

TABLE 2. UNITS OF CAPACITY

16 **Steam Boilers and Superheaters.*** (a) Heat output in steam per hour. (Pars. 113, 130.) (b) Actual evaporation, lb. of steam per hour, at stated steam pressure and quality or temperature, and stated feedwater temperature. (c) Units of evaporation per hour =Item 16a/1000. (Par. 113.)

17 **Reciprocating Steam Engines.** (a) Indicated horsepower at stated conditions of steam supply and exhaust. (b) Brake horsepower at stated conditions of steam supply and exhaust.

18 **Steam-Engine Generators.** Net kilowatts at generator terminals at stated conditions of steam supply and exhaust. (Par. 114.)

19 **Steam Turbines.** Brake horsepower at stated conditions of steam supply and exhaust.

20 **Turbo-Generators.** Net kilowatts at stated conditions of steam supply and exhaust. (Par. 114.)

21 **Pumping Machinery.** (a) Gallons discharged in 24 hours at stated total suction and discharge pressures. (b) Gallons per minute at stated total suction and discharge pressures. (c) Water-horsepower output at stated total suction and discharge pressures. (Par. 119.)

22 **Compressors and Blowers' Centrifugal and Displacement.** (a) Cubic feet of free air (or other, gas) per minute, at stated total intake pressure and temperature delivered at stated total discharge pressure. **Low-Pressure Centrifugal only**

* See comments at end of this appendix.

(less than 20 in. water total pressure rise). (b) Cubic feet of free air (or other gas) per minute, at one standard atmosphere or at 0.075 lb. per cu. ft., standard density for air, delivered at stated static discharge pressure. (c) Air-horsepower output at stated inlet and delivery conditions. (Par. 120.)

24 **Gas Producers.** (a) Pounds of fuel as fired per hour, of stated high calorific value. (Par. 121.) (b) Hot-gas output: cu. ft. of dry gas per hour, at stated temperature and pressure, and stated high calorific value. (c) Cold-gas output: cu. ft. of dry gas per hour at 68 deg. fahr. and one standard atmosphere. (d) Heat output per hour in hot gas. (e) Heat output per hour in cold gas.

25 **Gas and Oil Engines.** (a) Brake horsepower. (b) Indicated horsepower. (Par. 115.)

26 **Hydraulic Turbines.** Brake horsepower.

27 **Hydraulic Turbo-Generators.** Net kilowatts at the generator terminals. (Par. 114.)

28 **Condensers.** Heat transferred per hour, at stated vacuum, inlet and outlet circulating-water temperatures, and cubic feet of gases discharged by air pump per hour, measured at 68 deg. fahr. and one standard atmosphere. (Pars. 122–125.)

30 **Feedwater Heaters and Fuel-Oil Heaters.** Heat transferred per hour at stated steam pressure and temperature and at stated inlet and outlet water or oil temperatures. (Pars. 122–128.)

32 **Economizers.** Heat transferred per hour at stated pounds of flue gases per hour, inlet gas temperature and inlet and outlet water temperatures. (Pars. 122–128.)

33 **Cooling Towers and Cooling Ponds.** Heat dissipated per hour at stated inlet and outlet water temperatures, stated air temperatures and humidity.

34 **Refrigerating Machines.** (a) Heat absorbed per hour at stated head pressure and stated suction or cooler pressure. (Par. 122.) (b) Standard ton of refrigeration per 24 hrs. at stated head pressure, and stated suction or cooler pressure. (Par. 129.)

TABLE 3. UNITS OF PERFORMANCE

35 **Boilers (including firing equipment and superheaters).** (a) Efficiency of boiler, superheater and furnace: ratio of heat units output to high calorific value of fuel as fired. **Solid Fuels only.** (b) Rate of combustion in lb. of fuel as fired per sq. ft. of grate surface. per hour (Par. 131.) **All Fuels.** (c) Rate of combustion in lb. of fuel as fired per cu. ft. of furnace volume. (Par. 132.) (d) Heat transferred per sq. ft. of heating surface per hour. (Par. 122.) (e) Heat developed per sq. ft. of grate surface per hour. (Par. 131.)

36 **Reciprocating Engines.** (a) Heat supplied per i.hp-hr., b.hp-hr. (Pars. 140, 141.) (b) Thermal efficiency referred to i.hp., b.hp. (Pars. 134–136.) (c) Rankine efficiency referred to i.hp., b.hp. (Pars. 144–147.) (d) Water rate, pounds of steam per i.hp-hr., b.hp-hr. (Par. 152.) (e) Mechanical efficiency.

37 **Steam Turbines.** (a) Heat supplied per b.hp-hr. (Pars, 140, 141.) (b) Thermal efficiency. (Pars. 134–136.) (c) Rankine efficiency. (Pars. 144–147.) (d) Water rate, pounds of steam per b.hp-hr. (Par. 152.)

38 **Steam-Engine Generators and Turbo-Generators.** (a) Heat supplied per net kw-hr. (Pars. 114, 140, 141.) (b) Thermal efficiency. (Pars. 134–136.) (c) Rankine efficiency referred to net kw. (Pars. 144, 147.) (d) Water rate, lb. of steam per net kw-hr. (Pars. 114–152.)

39 **Steam-Driven Pumping Engines.** (a) Heat supplied per water hp-hr. (Pars. 140, 141.) (b) Thermal efficiency (Pars. 134–136) referred to water hp. (c) water rate, lb. of steam per water hp-hr. (Par. 152.) (d) Mechanical efficiency.

40 **Steam-Driven Compressors' Blowers and Fans.** (a) Heat supplied per air hp-hr. (Pars. 120, 140, 141.) (b) Thermal efficiency. (Pars. 134–136.) (c) Water rate, lb. of steam per air hp-hr. (Par. 152.) (d) Gross, (e) Net, (f) Indicated horsepower per cu. ft. dry free air per min. (g) Volumetric efficiency, for displacement compressors only. (h) Compression efficiency. (i) Mechanical efficiency.

41 **Direct-Drive Steam Plants.** (a) Fuel rate, lb. of fuel as fired per i.hp-hr., b.hp-hr. (Par. 154.) (b) Water rate, lb. of steam generated per i.hp-hr., b.hp-hr. (Par. 152.) (c) Heat in fuel, per i.hp-hr., b.hp-hr. (Pars. 140, 141.) (d) Thermal efficiency referred to i.hp., b.hp. (Pars. 134, 138.)

42 **Steam-Electric Plants.** (a) Fuel rate, lb. of fuel as fired per net kw-hr. (Pars. 114, 154.) (b) Water rate, lb. of steam generated per net kw-hr. (Par. 148.) (c) Heat in fuel per net kw-hr. (Pars. 114–116 and 140, 141.) (d) Thermal efficiency, overall, referred to net kw. (Pars. 132–138.)

43 **Steam Pumping Plants.** (a) Fuel rate, lb. of fuel as fired, per water hp-hr. (Pars. 119, 154.) (b) Water rate, lb. of steam generated per water hp-hr. (Pars. 119, 148.) *(c) Heat in fuel per water hp-hr. (Pars. 119, 140, 141.) (d) Thermal efficiency, referred to water hp. (Pars. 119, 132–138.)

44 **Steam Air-Machinery Plants.** (a) Fuel rate, lb. of fuel per air hp-hr. (Pars. 120, 154.) (b) Water rate, lb. of steam generated per air hp-hr. (Pars. 120, 152.) (c) Heat in fuel, per air hp-hr. (Pars. 120, 141.) (d) Thermal efficiency, referred to air hp. (Pars. 120, 134, 138.)

46 **Gas Producers.** (a) Hot-gas efficiency; ratio of high calorific value plus sensible heat above room temp. in hot gas, to the high calorific value of fuel as fired. (Par. 121.) (b) Cold-gas efficiency; ratio of high calorific value of gas to high calorific value of fuel as fired. (Par. 121.)

47 **Internal-Combustion Engines.** (a) Fuel rate, lb. of fuel as burned per net i.hp-hr., net b.hp-hr. (Par. 154.) (b) Fuel rate, cu. ft. of dry gas of stated high cal. value at 68 deg. fahr. and one standard atmosphere, per i.hp-hr., b.hp-hr. (c) Heat supplied per i.hp-hr., b.hp-hr. (Pars. 140, 141.) (d) Thermal efficiency referred to i.hp-hr., b.hp-hr. (e) Otto or Brayton efficiency referred to i.hp-hr., b.hp-hr. (Pars. 144, 149–151.)

48 **Gas or Oil-Electric-Units.** (a) Fuel rate, lb. of fuel as burned per net kw. (Pars. 154, 114, 116.) (b) Fuel rate, cu. ft. of dry gas of stated high cal. value at 68 deg. fahr. and one standard atmosphere per net kw-hr. (c) Heat supplied per net kw-hr. (Pars. 140, 141.) (d) Thermal efficiency. (Pars. 134–137.) (e) Otto or Brayton efficiency referred to net kw. (Pars. 144, 149–151.)

49 **Gas-Producer Plants.** (a) Fuel rate, lb. of fuel as fired per net i.hp-hr., b.hp-hr. (Pars. 115, 154.) (b) Heat in fuel per net i.hp-hr., b.hp-hr. (Pars. 140, 141.) (c) Thermal efficiency referred to i.hp-hr., b.hp-hr. (Pars. 134–137.)

50 **Hydraulic Turbines.** Efficiency of turbine; ratio of brake horsepower to water horsepower. (Par. 119.)

51 **Hydraulic Turbo-Generators.** Efficiency of unit; ratio of net electrical hp. to water hp. (Pars. 114–119.)

52 **Surface Condensers.** (a) Heat transferred per sq. ft. of cooling surface under stated conditions of vacuum, inlet and outlet circulating-water temperatures and cu. ft. of gases discharged per hour, at 68 deg. fahr. and one atmosphere. (Pars. 123, 124.) (b) Heat-transmission coefficient. (Par. 157.)

54 **Closed Feedwater Heaters and Fuel-Oil Heaters.** (a) Heat transferred per sq. ft. of heating surface per hour at stated conditions of steam supply, and stated inlet and outlet temperatures of water or oil. (Par. 128.) (b) Heat-transmission coefficient.

56 **Economizers.** (a) Heat transferred per sq. ft. of heating surface per hour at stated lb. of flue gases per hour, inlet flue-gas temperature and inlet and outlet water temperatures. (Par. 128.) (b) Heat-transmission coefficient. (Par. 157.)

57 **Cooling Towers.** (a) Efficiency. (Par. 158.)

58 **Steam-Driven Refrigerating Machines.** (a) Heat supplied in steam per ton of refrigeration, at stated conditions. (Pars. 129, 154.) (b) Coefficient of performance; ratio of heat abstracted to indicated work, expressed in B.t.u. (c) Water rate, lb. of steam per ton of refrigeration, at stated conditions. (Pars. 129, 148.)

* See comments at end of this appendix.

Notes on Data

101 In order to render the terminology of the Codes consistent, the following symbols in general use are adopted:

H = heat content of the vapor (Par. 111)
h = heat content of the liquid (Par. 111)
L = latent heat, or heat of vaporization
C_p = specific heat at constant pressure
C_v = specific heat at constant volume
$\gamma = \frac{C_p}{C_v}$
v = specific volume of the vapor cu. ft. per lb.
v' = specific volume of the liquid cu. ft. per lb.
E = efficiency
I = Internal energy
p = absolute pressure
N = entropy of vapor
n = entropy of liquid
J = mechanical equivalent of heat
A = heat equivalent of work $= 1/J$
T = absolute temperature
t = temperature, deg. fahr.
V = velocity, ft. per sec.
W = work
w = weight
Q = quantity of heat in general; not to be used for denoting the property of a particular substance in a particular state.

102 Congress has never fully exercised its constitutional power of fixing the standards of weights and measures throughout the United States. The Bureau of Standards is now the legal custodian of the standards; and in the absence of specific action to the contrary by Congress, its practice in determining the values of lengths and masses is authoritative.

103 The ultimate standard of length is the international prototype meter, of which the Bureau has the two official copies assigned by lot to the United States. Determinations of length are referred to the ultimate standard through these copies by using the relation 1 meter = 39.37 in. Similarly, the ultimate standard of mass is the international prototype kilogram, of which two official copies are in the custody of the Bureau of Standards; and determinations of mass are referred to the kilogram by using the relation 1 lb. = 0.4535924 kg. There is no legally established United States primary standard yard bar or pound mass, and the adoption of the numerical relations mentioned above is, in effect. a definition of the foot and the pound.

104 For further information, see The History of the Standard Weights and Measures of the United States, by L. A. Fischer, *Bulletin* of the Bureau of Standards, Vol. 1, p. 365 (1905).

105 *International Standard Gravity*, adopted in 1901 by the committee of the International Bureau of Weights and Measures, has the value

$$g = 980.665 \text{ cm./sec.}^2 = 32.1740 \text{ ft./sec.}^2$$

which is the actual value of this acceleration at sea level and about 45 deg. latitude. At other latitudes, and at sea level, the ratio of local to standard gravity is as shown in the following table:

Latitude, (deg.)	$\frac{g \text{ (local)}}{g \text{ (standard)}}$	Latitude, (deg.)	$\frac{g \text{ (local)}}{g \text{ (standard)}}$
0	0.9973	50	1.0004
10	0.9975	60	1.0013
20	0.9979	70	1.0020
30	0.9986	80	1.0024
40	0.9995	90	1.0026

For higher altitudes, subtract 1 part in 10,000 for each 1000 ft. above sea level. At an elevation of 10,000 feet at the equator, local gravity is about 1 part in 79 less than standard gravity. For latitudes between 20 deg. and 70 deg. and altitudes below

5000 ft., the maximum difference between local and standard gravity is about 1 part in 400, an amount which is ordinarily negligible for engineering purposes.

106 For the mechanical equivalent of heat, the value

$$J = 778 \text{ ft-lb. per B.t.u.}$$

is adopted: it is equivalent to 1 mean calory = 4.186 true joules.

Other values which have been used in some of the recent steam tables are: Mollier (1906), 778.28; Marks and Davis (1916), 777.52; Goodenough (1914), 777.64; Callendar* (1915), 777.8. The greatest difference of any of these values from 778.00 is about 1 in 1620, an amount which is devoid of physical significance because the uncertainty of the value is at least 1 part in 500 and perhaps 1 in 200. The 15 deg. calory is known to about 1 part in 2000, and the mean calory is known to be nearly the same as the 15-deg. calory. But the difference between the two is not so accurately known, even its sign being uncertain. In view of this fact it is quite evident that in the value $J = 778$ ft.-lb. per mean B.t.u., even the 8 is not certain, and that the use of additional figures beyond the decimal point is a purely illusory refinement.

107 With $J = 778$ ft-lb. per B.t.u. we have

$$1 \text{ horsepower-hour} = \frac{1,980,000}{778} = 2544.987 + \text{B.t.u.}$$

$$1 \text{ kilowatt-hour} = \frac{1,980,000}{778 \times 0.745702} = 3412.874 + \text{B.t.u.}$$

Since the value 778 is uncertain by more than one unit, the use of decimal places in the foregoing values is a pure waste of time and they are abbreviated to 2545 and 3413.

108 *Absolute Temperature* in Fahrenheit degrees is denoted by

* Callendar's fundamental value is 1 pound-degree centigrade = 1400.00 London foot-pounds, the British foot and pound mass being sensibly identical with the U. S. values, while the value of gravity at London is $g = 981.19$ cm./sec.^{2}.

$T°$, and temperature on the ordinary Fahrenheit scale by $t°$, the relation between the two being

$$T° = t° + 459.7$$

and the absolute temperature of the ice point being $T° = 491.7$. The values are uncertain by about two units in the last place given. For all ordinary engineering purposes, such as reductions of gas volumes, the values 460 and 492 are more than sufficiently accurate.

109 *Ice Point and Steam Point.* The ice point is 32 deg. Fahr. and the steam point 212 deg. Fahr., both at one standard atmosphere.

110 *Internal Energy of a Substance.* When a pound of substance is brought from one state to another, for example, when a pound of water at 100 deg. Fahr. and 50 lb. per sq. in. pressure is changed into steam at 100 lb. per sq. in. pressure and 70 deg. superheat, the amount of heat required for the change is not definite but depends altogether on the "path" of the change, i.e., on the series of intermediate states. But the sum of the heat put into, and the heat equivalent of the work done on the pound of substance during the change of state is definite and depends only on the initial and final states and not on the path of the change. This sum is the value, expressed in heat units, of the energy that must be added to the pound of substance to produce the change, or it is the increase of the *internal energy*, in B.t.u. per lb., during the change. Since we are always concerned with *changes* of internal energy and not with absolute values, the internal energy may be set arbitrarily equal to zero at any convenient standard or normal state, such as 32 deg. Fahr. and one atmosphere pressure. When such a convention has been adopted, the value of the internal energy in any other state is thereby fixed; and if the necessary data are available, it may be tabulated. Internal energy is denoted by the symbol I and expressed in mean B.t.u. per lb. of substance.

111 *Heat Content.* Let v' be the volume in cubic feet, and I the internal energy in B.t.u. of one pound of a fluid at the pressure p in lb. per sq. in. Then the important quantity

$$H=I+\frac{144pv'}{J}=I+144\ Apv' \text{ B.t.u. per lb}$$

has been called the "total heat," "heat of formation" or "heat content" of the fluid, no one of the names being very satisfactory. The first is the most usual; but the necessity of using the adjective "total" in its ordinary sense sometimes makes it difficult to avoid ambiguity when the word is also being used with a special technical meaning in the compound noun "total heat." The name "heat content" will therefore be used for the quantity denoted above by H. When the substance is all in the liquid state, the symbol h will be used instead of H, and h thus denotes what is commonly called the "heat of the liquid."

112 *The standard density of mercury* at 32 deg. fahr. is 13.5955 grams per cu. cm. (Kaye & Laby, 1918). The mean cubical expansion between 32 deg. Fahr. and 212 deg. Fahr. is 0.0001014 per deg., and from 32 deg. to 110 deg. Fahr. it is 0.0001010.

113 *Units of Evaporation.* If it is desired to reduce B.t.u. output per hour to *units of evaporation,* divide by 1000. The above changes are made for the following reasons: The boiler horsepower as originally standardized by the A.S.M.E. in 1889 was based on a conventional engine water rate of 30 lb. of steam per hp-hr. at 70 lb. gage pressure and feedwater at 100 deg. Fahr. This corresponds to 34.5 lb. evaporated from and at 212 deg. Fahr. (33479 B.t.u. per hr.) At the present time, water rates vary from 8 lb. per b.hp. hr. in large condensing turbines to 50 or 60 lb. for small non-condensing units, and the boiler horsepower has no connection whatever with the water rate of the engine. It has never been used on the Continent or in England, and in the United States marine boilers are rated in horsepower according to the hp. and water

rate of the engines they serve. The unit has therefore become archaic, and it is better to state capacity in B.t.u., or in multiples of B.t.u. Under the boiler-horsepower definition, the unit of evaporation is the latent heat of vaporization at 212 deg., the value for which has varied with the steam tables from 965 to over 970 B.t.u. per lb. The statement of capacity or performance in B.t.u. is basic, and as the computations must in any case be made in B.t.u. at some point in the calculations, it is convenient to omit other arbitrary units as unnecessary. The objection that has been raised that capacity in B.t.u. gives large numbers is hardly worth consideration; it is only necessary to head the columns "units of evaporation" (1000 B.t.u.) which is but three per cent different from "equivalent evaporation" as now used, or "million B.t.u.," in which case the figures are smaller than when stated in boiler hp. The rating of boilers for size only, in square feet of heating surface, should be adopted; the area method has been in use for years in Europe, and in view of the fact that the ultimate steaming capacity per square foot is dependent solely on amount of fuel fired, it is a suitable unit, as not involving capacity.

114 *Net Output.* For any kind of separately excited engine or generator, the net output is expressed by the following formula:

Net kw. = gross kw. (main unit)—kw. excitation at collector rings

Net output where direct connected exciters are employed, is expressed by the following formula:

Net kw. = gross kw. (main unit) plus gross kw. exciter—kw. excitation at collector rings.

The gross kw. of both main unit and exciter is to be measured at the generator terminals. Further correction must be made if separately driven ventilating fans are employed, by substituting kw. to fan motor, or as determined by prior agreement, if the fan is not motor-driven.

115 *Net I.Hp.* for internal combustion engines, is the i.hp. of main cylinders minus the i.hp. of auxiliary cylinders for scavenging and injection. Where possible the net output should be in b.hp., eliminating all corrections. I.hp. is, strictly speaking, not a true output, inasmuch as it does not represent the power available for use at the engine shaft.

116 For complete power stations, gross output is the sum of the gross outputs of individual units. The difference between gross and net outputs is the kw. used for lighting, auxiliaries and house service and has the same character (necessary loss) for the complete plant as friction or excitation for an individual unit. It will be seen the term "net" is used to indicate that output which is available for use, outside the main unit, or plant, as the case may be.

117 *Rate of Combustion* is defined in two ways; the first is confined to solid fuels. For this case it is the pounds of fuel as fired per square foot of grate surface per hour.

118 Rate of combustion for the second case is used for all fuels, and comprises the pounds of fuel as fired per cubic foor of furnace volume per hour. For gas only, it may be stated as the cubic feet of gas as fired per cubic foot of furnace volume per hour.

119 Water horsepower is to be computed from the equation—

$$\text{Water hp} = \frac{(\text{lb. of liquid per min.}) \times (\text{total head in ft.})}{33{,}000}$$

Both suction and discharge heads are to be total heads, as given by the impact tube, except in cases where the velocity head is less than 0.2 per cent of the total head. In the latter case static pressure, as usually taken by pressure gages, may be used.

If the total head is given as a difference of pressure, the value to be used in the foregoing equation is to be found from the formula

$$\text{Head in feet} = 144 \times \frac{\text{Pressure difference in lb. per sq. in.}}{\text{Actual density of water in lb. per cu. ft.}}$$

Water horsepower for pumps is the energy supplied to the water or other liquid by the pump in unit time. In the case of hydraulic turbines it is the available energy in the water to be employed by the turbine in unit time.

120 *Air Horsepower** (air hp.) is defined as the horsepower that would be required to compress the actual air output of the compressor or blower, if there were no friction and no clearance, if the inlet and outlet pressures were constant, and if the compression were adiabatic, from the temperature at intake.

The suction and discharge pressures must be the total pressures as obtained with the impact tube, so as to include velocity head, except in those cases where the difference of velocity head at inlet and discharge is less that 0.2 per cent of the total head, in which case the usual static pressures given by pressure gages may be used.

121 *Calorific Value* as used in the tables, for solid and liquid fuels, is in all cases the high heat value per pound on complete combustion. The calorific value for gas is the high heat value per cubic foot. The low heat value is not to be used. The standardization on high heat values is adopted in order that all heat apparatus shall be charged with heat supplied, on the same basis. If the high heat values are used for some heat engines, and the low values for others, the efficiencies and other performance figures will not be comparable.

122 *B.t.u. Transferred per Hour.* In heat-transfer apparatus heat is transferred, or flows, from a region of higher to a region of lower temperature, without change of total quantity. The heat is nearly always given up or received by a fluid, and the rate of transfer is determined by measuring the rate at which the fluid gains or loses heat. Usually, as in surface condensers, evaporators, etc., both sides of the apparatus contain fluid, and measurements may be made on either of the fluids. If the apparatus is well

* See comments at end of Appendix B.

insulated or is at nearly the same temperature as its surroundings so that the external heat transfer is negligible, the rate will be the same in which ever way it is found, but different methods may be most suitable in different cases.

123 *Surface Condensers.* Let w_c = total pounds of cooling water per hour and let t_1 and t_2 = its mean inlet and outlet temperatures. Then the rate of heat transfer is—

$$Q = w_c(t_2 - t_1) \text{ B.t.u. per hr.}$$

this being the rate at which the water receives heat.

124 It will usually be more convenient to make the measurement on the steam side. Let w_s = total pounds of steam condensed per hour; let H_2 = heat content, in B.t.u. per lb., of the steam as it enters the condenser; and let h_2 = heat content in B.t.u. per lb. of the water in the hot well. Then the rate of heat transfer from the steam is—

$$Q = w_s(H_2 - h_2) \text{ B.t.u. per hr}$$

The "heat of the liquid" h_2 may be found from the steam table if the back pressure is known, but the value of H_2 has to be determined indirectly because the dryness factor of the steam cannot well be observed, and H_2 can therefore not be found directly from the steam table.

125 *The value of w_sH_2 is to be found from the equation

$$w_sH_2 = w_sH_1 - A - R$$

where H_1 = heat content of initial steam from the steam table

A = extraction in B.t.u. per hr.

R = heat lost by the engine to the surroundings in B.t.u. per hr. (commonly but incorrectly called "radiation loss").

a) For reciprocating engines—

$$w_sH_2 = w_sH_1 - \text{i.hp.} \times 2545 - R$$

$$= w_sH_1 - \frac{\text{b.hp.} \times 2545}{\text{mech. effy.}} - R$$

* See comments at end of Appendix B.

(*b*) For steam turbines—

$$w_s H_2 = w_s H_1 - \text{b.hp.} \times 2545 - R$$

(*c*) For turbo-generators—

$$w_s H_2 = w_s H_1 - \frac{\text{kw} \times 3413}{\text{generator effy.}} - R$$

Except for small turbines the heat loss R and the bearing and gland friction are negligible: all other losses appear as reheat in the steam.

128 *Feedwater Heaters, Fuel-Oil Heaters, Coolers and Economizers.* The rate of heat transfer is—

$$Q = w_c c(t_2 - t_1) \text{ B.t.u per hr.}$$

where w_c = total pounds of water or oil passed through per hour,
c = the specific heat of the substance (for water, $c = 1$)
$(t_2 - t_1)$ = the rise or fall of temperature.

129 *One Standard Ton of Refrigeration* is defined as the absorption of 288,000 B.t.u. irrespective of time. This definition does not agree exactly with the heat of fusion of ice according to the latest determinations but has been adopted as a conventional standard by both the A.S.R.E. and the A.S.M.E.

The unit of capacity, one ton per day, will then be $\frac{288{,}000}{24 \times 60} = 200$ B.t.u. per min.

130 *Total B.t.u. per Hour Output in Steam from Boiler:*

$$Q = W(H_1 - h_2) \text{ B.t.u. per hr.}$$

where W = total pounds of water evaporated per hour
H_1 = heat content of the steam generated in B.t.u. per lb.
h_2 = heat content of feedwater in B. t.u. per lb.
H_1 and h_2 to be found from the steam table.

131 *Grate Surface** is defined as the total horizontal projected area of grates or stoker, including dump plates, ash crushers, etc.

* This definition is to be revised and extended.

It is also stated as the total projected area of all surface supporting coal, within the front wall of the furnace. This definition will cover cases in which the bridge wall is undercut for ejection of refuse.

132 *Total Furnace Volume* is defined for horizontal return tubular boilers and water tube boilers as the cubical contents of the furnace between the grate and the first place of entry into or between tubes. It therefore includes the volume behind the bridge wall as in ordinary horizontal return tubular boiler settings, unless manifestly ineffective (i.e., no gas flow taking place through it), as in the case of waste-heat boilers with auxiliary coal furnaces, where one part of the furnace is out of action when the other is being used. For Scotch or other internally fired boilers it is the cubical contents of the furnace, flues and combustion chamber, up to the plane of first entry into the tubes.

133 *Heating Surface* for boilers comprises the total area of surface in actual contact with hot gas and below the normal water level of the boiler, provided the heating surface comprises a part of the circulation system of the boiler proper. If any such surface is not in the boiler circulation system, and is not connected to the steam space of the boiler, it is to be considered as preheater, or integral economizer surface, and not as boiler heating surface. Superheater surface is the total area of all surface in contact with the hot gases. Superheater, boiler, and preheater surface should be separately stated. Since the gas side of the surface offers the controlling resistance to heat transmission, the surface will be figured on the outside diameter of tubes for water tube boilers, and on the inside diameter for fire tube boilers.

Heating or cooling surface, for condensers, evaporators, feed-water heaters, oil heaters and oil coolers, will be figured on total surface in contact with both fluids, and based on outside diameter of tubes. Heating surface for economizers will be figured the same as for boilers. Heat transfer should be figured separately for preheaters, boilers and superheaters, based on the surface in-

volved and the actual changes of heat content between entry and delivery of each.

134 *Thermal Efficiency* based on any unit of output is defined as the heat equivalent of the work done divided by the heat supplied.

135 *The Heat of the Liquid,* for steam engines and turbines, is to be taken at the temperature corresponding to the back pressure.

136 *Thermal Efficiency* is expressed as follows:

For steam engines:

$$\text{Indicated thermal efficiency, } E_i = \frac{2545}{w_i(H_1 - h_2)}$$

For steam engines and turbines:

$$\text{Brake thermal efficiency, } E_b = \frac{2545}{w_b(H_1 - h_2)}$$

For engine-generators and turbo-generators:

$$\text{Combined thermal efficiency, } E_k = \frac{3413}{w_k(H_1 - h_2)}$$

where w_i = steam consumption referred to indicated horsepower
w_b = steam consumption referred to brake horsepower
w_k = steam consumption referred to net kilowatts
H_1 = heat content, B.t.u. per lb. at the throttle
h_2 = heat of the liquid, B.t.u. per lb., corresponding to pressure in exhaust.

137 *The Thermal Efficiencies* of the internal-combustion engine are expressed as follows:

$$\text{Indicated thermal efficiency, } E_i = \frac{2545}{Q_i}$$

$$\text{Brake thermal efficiency, } E_b = \frac{2545}{Q_b}$$

$$\text{Combined thermal efficiency, } E_k = \frac{3413}{Q_k}$$

where Q_i = B.t.u. per i.hp.
Q_b = B.t.u. per b.hp.
Q_k = B.t.u. per net kw.

138 The thermal efficiency for complete plants will be expressed in the same way, using i.hp., b.hp., net kw., water hp., air hp., etc., as the reference. For example, the *overall thermal efficiency* of a coal-fired electric plant is—

$$\frac{3413}{\text{Calorific value of coal} \times \text{lb. coal per net kw-hr.}}$$

139 It is to be noted that in present usage the terms "thermal efficiency" and "cycle" are not limited to the thermodynamic sense only. As used in the Code, they are therefore not to be interpreted in the strictly special manner usual in thermodynamics. Their use in the general sense is established by custom and is perfectly understandable.

140 *Heat Supplied*, referred to any unit of output, is defined as the heat input per unit of output.

141 *Heat Supplied*, for steam engines and turbines, is expressed as the total heat content of the steam supplied less the heat of the liquid at exhaust pressure. For complete steam plants, for gas producers, internal-combustion engines and internal-combustion plants, it is expressed as the high calorific value of the fuel per lb. as fired, times the pounds fired.

142 *The Initial Steam Pressure* for any steam engine or turbine is defined as the average pressure obtained in the supply pipe directly preceding the throttle valve of the engine or turbine. The same definition will apply to any other apparatus using steam and using a stop valve to start or stop it.

143 *The Back Pressure or Exhaust Pressure* for steam engines or turbines is defined as the pressure obtained at or as near as possible to the exhaust flange, and it shall also be considered to be the pressure obtaining in the condenser, where the condenser is directly connected to the turbine or engine-exhaust flange (applied only to steam prime movers).

144 *Engine Efficiency* is the general term used for the ratio between heat input per unit of output for the ideal cycle, and heat

input per unit of output for the actual engine. It may also be expressed as $\frac{E_i}{E_r}$, $\frac{E_b}{E_r}$ or $\frac{E_k}{E_r}$; and in other ways; all of which are simply transpositions of the same quantities. The ideal cycles are different for the different classes of prime movers. For steam engines and steam turbines, the ideal is the Rankine cycle, and the above ratio will be called the "Rankine efficiency." For explosion internal-combustion motors the Otto cycle is the ideal, and the ratio will be called the "Otto efficiency." For constant-pressure internal-combustion motors the Brayton cycle is the ideal, and the ratio will be called the "Brayton efficiency." Other cycles may be employed as ideals for comparison, as prime movers are developed, but are not required at present. The Carnot cycle, although it affords the highest thermal efficiency, is not employed in any commercial prime mover at present, and therefore is of no practical value in these codes.

145 *The Rankin Steam Cycle* consists of (*a*) admission at constant pressure and temperature, (*b*) isentropic expansion to the back pressure, (*c*) exhaust at constant pressure and temperature, (*d*) return to the boiler of the equivalent amount of feedwater, taken at the temperature and pressure of the exhaust steam; there is to be no heat leakage and no friction, so that all stages of the cycle are ideally perfect.

146 *Thermal Efficiency of the Rankine Cycle:*

$$E_r = \frac{H_1 - H_2}{H_1 - h_2}$$

where H_1 = heat content of steam, at initial condition
H_2 = heat content of steam after isentropic expansion
h_2 = heat of the liquid at exhaust pressure.

The foregoing formula is not exact because it neglects the work of the feed pump, but this is in fact negligible, so that the formula may be used without correction.

147 *Heat Input of the Rankine Cycle* is expressed as

$$\frac{(H_1-h_2)\,2545}{H_1-H_2}=\frac{2545}{E_i} \text{ for i.hp.}$$

$$\frac{(H_1-h_2)\,2545}{H_1-H_2}=\frac{2545}{E_b} \text{ for b.hp.}$$

$$\frac{(H_1-h_2)\,3413}{H_1-H_2}=\frac{3413}{E_k} \text{ for net kw.}$$

148 *The Otto Cycle* consists of (*a*) adiabatic compression, (*b*) heating at constant volume (explosion), (*c*) adiabatic expansion to the original volume, and (*d*) cooling at constant volume (exhaust).

149 *The Brayton Cycle* (of which the Diesel cycle is a modification), is defined as (*a*) adiabatic compression, (*b*) heating at constant pressure, (*c*) adiabatic expansion to the back pressure, and (*d*) exhaust at constant pressure.

150 *Air Standard Thermal Efficiency of the Otto and Brayton Cycles* is expressed as

$$E_o=1-\left(\frac{p_a}{p_b}\right)^{\frac{\gamma-1}{\gamma}}=1-\left(\frac{v_b}{v_a}\right)^{\gamma-1}$$

where E_o = thermal efficiency of the Otto or Brayton cycle.

p_a and v_a = absolute pressure and volume at beginning of compression.

p_b and v_b = absolute pressure and volume at end of compression.

γ = ratio of specific heats, $\frac{C_p}{C_v}$ (equals 1.40 when air is used in compressive gas cycles)

The value of γ in the foregoing equation is taken at 1.40, from the value 1.402, for air at the ice point and one atmosphere; the value 1.41 heretofore employed is too high. Even the value herein assumed is purely conventional, inasmuch as γ is not constant; the average value γ from room temperature to 4000 deg. Fahr.

and one atmosphere is 1.36, and at higher pressure is probably even lower. The air cycle with constant value of γ is about the only present manageable standard, although it is recognized that it gives Brayton and Otto efficiencies that are somewhat too low.

151 *Heat Input for Otto and Brayton Cycles* is expressed as

$$\text{B.t.u. per i.hp-hr. or b.hp-hr.} = \frac{2545}{E_o}$$

$$\text{B.t.u. per net kw-hr.} = \frac{3413}{E_o}$$

152 *Water Rate* of an engine, turbine, or complete steam plant is the pounds of steam at actual condition, per unit of output; it is to be corrected neither for moisture nor superheat.

153 *The Quality of Steam* or other vapor is specified in two ways: If the vapor is superheated, the quality is described by stating the number of degrees of superheat. If the vapor is wet, the quality is described by stating the dryness factor; for example, if steam contains 2½ per cent of moisture, its dryness is 0.975.

Dry Steam, to which most steam conditions involving wet steam were formerly corrected, is that condition at which there is present no moisture and no superheat, and the heat content is that of the point of saturation. It is a condition which cannot be exactly realized commercially, and as the correction to dry steam has never been applied to superheated steam, it appears advisable to discontinue its use. This does not bar its employment, as a condition, together with a definite pressure (and back pressure or vacuum), for correcting guarantee tests to a common state.

154 *Fuel Rate,* for solid and liquid fuels, is defined as the pounds of fuel as fired, per unit of output per hour. For gaseous fuels it is defined as cubic feet of gas at 68 deg. Fahr.* and one atmosphere, per unit of output per hour. It should be qualified by reference to the unit of output.

* See comments at end of this Appendix.

157 *Heat-Transmission Coefficient* is defined as the British thermal units transmitted per square foot of heating surface per degree mean temperature difference per hour.

158 *Cooling-Tower Efficiency* is expressed by the formula—

$$\text{Efficiency} = \frac{t_1 - t_2}{t_1 - t_w}$$

where t_1 = temperature of water before cooling
t_2 = temperature of water after cooling
t_w = temperature of wet bulb.

(*Signed*) REGINALD J. S. PIGOTT, *Chairman*,
EDGAR BUCKINGHAM,
FRED R. LOW,
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COMMENTS ON THE CODE *

Heat Content of Steam. This is defined in paragraphs 110 and 111 of the code. These paragraphs should be understood, since the term "Heat content" is frequently used elsewhere in the code. The notation is H with subscripts to signify different conditions. According to the paragraphs mentioned H stands for heat content of wet, dry, or superheated steam, and has the same meaning as on page 140 of this book. In former Power Test Codes H stood for the heat content of saturated steam. In tests of steam engines and boilers the results were "corrected for moisture" in a roundabout way, to allow for the fact that H for one lb. of wet steam equals $h+xL$, instead of $h+L$. It is to be regretted that the Boiler Test Code Committee, in its latest report, adheres to this custom, despite the recommendations of the Definitions and Values Code.

* Made by J. C. S.

Standard Cubic Feet of Air or Gas. Items 22, 24(c), 47, 48, etc. This is given at 68 deg. F. instead of 32° as recommended elsewhere in this book. The latter is the scientific standard and is to be preferred on that account. The latest report of the Boiler Test Code Committee recommends the 32° standard for gas-fired boilers.

Steam Boilers. Items 16 and 35. The changes here sought are, perhaps, the most radical of any developed by the committee, and yet entirely logical. "Boiler Horse-power" is a misleading unit and "factor of evaporation" an unnecessary and, to the student, a confusing expression. The establishment of these standards is, however, still in doubt, because of the contrary action of the 1922 Boiler Test Code and because of the radical nature of the changes. It is possible that the term "Boiler Horse-power" will cling, especially among small manufacturers and users.

Steam Pumping Machinery. Item 43. It is to be noted that the "duty" rating is omitted. (a), (b), and (d) of item 43 may be used to compare with similar results of other prime movers. 43 (c) is inversely proportional to "duty" if duty is defined as foot-pounds of work done per million B.t.u. available in the fuel.

Par. 119. By "impact tube" is meant one so shaped that the opening inside the pipe faces the stream of fluid. This arrangement causes both velocity and pressure (or static) head to be recorded.

Air Horse-power. Items 22 and 40. Paragraph 120. The old standard of isothermal compression has been discarded here, and adiabatic compression used instead. This is illogical, for compressors with well jacketed cylinders, in cold weather, may easily reach an efficiency of compression of over 100 per cent if adiabatic compression be used as a basis. This suggested standard is subject to further change.

Heat Content of Exhaust Steam. Paras. 124 and 125. A numerical example illustrating this method is given on page 377. The notation used is somewhat different from that of the Code.

Engine Efficiency. Par. 144. This term is bad. It in no way signifies the meaning given under the definition immediately following. It is, however, a difficult matter to name a general quantity so that the name will at once imply the meaning, be short and euphonious, and also cover all of its specific applications. "Engine Efficiency" has the last two attributes, but not the first and most important. In the opinion of the writer this term will not come into general use.

These comments should not close with adverse criticism since the Code as a whole is so good. Its authors are to be congratulated. It is to be hoped that the other Power Code Committees will do everything possible to harmonize with this one.

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